700 Answers to Power Problems

700 Answers to Power Problems

Covering in simple question-and-answer style problems in every branch of power-practice. Based on questions submitted by puzzled engineers to the editorial staff of Power

Fourth Edition

Published 1929 by
POWER
Tenth Ave. at 36th St. New York, N. Y.

McGraw-Hill Publishing Co. Inc.
Tenth Ave. at 36th St.
New York, N. Y.

Preface

For many years, the Power Engineers of the country have been bringing their troubles to the editorial staff of Power, in the knowledge that the benefit of all its resources and facilities were unreservedly at the disposal of subscribers.

Into this book, we have put carefully worked out solutions to no less than 700 of the most frequently recurring of these troubles, in the belief that *every* power engineer is sure to find in this large compilation many practical helps in working out his everyday tasks.

You'll find the arrangement particularly convenient, as the grouping in fourteen sections and the comprehensive index make every fact instantly accessible.

Use this book often—it will pay you to do so.

Section I

Steam Turbines

QUESTION [Shutting Down Steam Turbine]—*In shutting down a steam turbine, is it advisable to throttle by hand before or after taking off the load?*

R. N. D.

Answer—It is advisable to throttle down as the load is taken off. Then, if the main valve leaks, overspeeding will be prevented. This is the usual custom with large units. When the governing mechanism is in good order, this may not be necessary and on small turbines with tight valves, closing the hand throttle is often delayed until the load is removed. Shutting down affords a good opportunity for tripping the emergency valve by hand to determine whether overspeeding occurs at no load, and to test the emergency, after which the throttle can be closed tight.

QUESTION [Noise in High-Speed Turbine Stuffing Box]—*The shaft gland of a small high-speed turbine that is provided with soft braided metallic packing, occasionally produces a slight popping noise. I tightened the gland slightly with the result that packing became very hot. What would you advise as the to cause of trouble?*

W. F. W.

Answer—The noise you mention is caused by drops of water coming through with the steam. These are sometimes re-evaporated by frictional heat and produce an explosive sound. This is perfectly normal and indicates no trouble, usually occurring when glands are tight and in good condition, and gradually disappearing in time.

If avoidable, the shaft packing should not be tightened while the machine is at full speed, as a slight excess of friction will quickly result in heating. Tightening should be done when the turbine is at low speed, so that tight spots can be worn out or forced into proper adjustment without causing excessive heating.

QUESTION [Sludging of Turbine Oil]—*We have a small non-condensing turbine operated at 100lb. gage against 4 lb. back pressure and have great trouble with oil sludging in summer, when our station load is heaviest, using the best oils obtainable. What would you advise as to the cause of our trouble?*

H. F.

Answer—Proba bly the bearings are overheated on account of the high back pressure of the exhaust, which has a tendency to heat the shaft. If the engine room is excessively hot in summer, as must be the case, the trouble may be due to overheating of the oil. If the temperature of the oil reservoir exceeds 180 deg. F., we suggest that you water-cool the oil during the hot season. Place a jointless coil in the reservoir of the turbine, or pipe the pressure line through a tight cooler. Make sure that no water, or at least very little, enters the oil system. If the shaft packing leaks badly, a great deal of water may be condensed inside the bearing housing.

QUESTION [Vibration of a Turbine Set Caused by Loose Joint in Generator Field]—*An alternating-current turbine set has run satisfactorily for several years. We recently reconnected the fields to operate on* 125-*volt exciting current, instead of* 250 *as formerly and lately have carried a heavy overload with low power factor, when vibration spells have developed. This is in the generator only. We cannot find any ground or other indications of electrical trouble, and the rotating field is aligned centrally with the armature. The set will run for hours perfectly when the generator is not in use, but vibration develops soon after load is applied. What causes the vibration?*

W. L. T.

Answer—The trouble appears to be due to a loose joint or terminal of one of the generator field coils, which under certain circumstances forms a make-and-break contact, causing a variable magnetic pull in one coil. Probably a terminal or joint was not soldered properly and the excessive field current carried on account of overload and low power factor caused a connection to give way. It sometimes happens that terminals become loose in this manner and are difficult to detect, since the trouble may occur only at certain load conditions when heating is sufficient to destroy proper mechanical contact. It may be possible to locate such a joint by exciting the field and taking the drop of potential across each joint with a multi-voltmeter. The bad joint will show an excessively large voltage drop when compared to the others.

QUESTION [Laying up Steam Turbines]—*We have two impulse turbines which may be operated condensing or non-condensing. Having ar anged to operate our plant by electric power receivd from a pu lic central station, what should be done in layin up the turbines to stand by in good condition, ready to start when needed?*

R. E. R.

Answer—When the turbines are shut down and drained, pour about one-half gallon of engine oil in the cylinder of each turbine when slowly turning over, and work the oil through

the cylinder to coat thoroughly the blading, packing glands and other finished parts. Care should be taken that the turbines are completely disconnected from any live or exhaus-steam. To guard against rust or damage from such connect tions, close the throttle valves and provide bleeders outside of the valves with independent openings to the atmosphere. Any drains connected with other drips should be disconnected close to the machines, and the drain pipes should be blanked off to prevent steam from backing up to the turbines. After the turbines are thoroughly drained, close the exhaust valves and open a bleeder outside of the exhaust valves to prevent any steam backing up from other connections. After all finished parts have received a doping of thick oil, provide the turbines with tarpaulins or canvas coverings for protecting them from dust.

QUESTION [Determining Erosion of Turbine Blading]—*How may it be known if the blading of a steam turbine is badly eroded without removing the casing?*

J. F.

Answer—The only way of telling whether the blading is eroded without taking the turbine apart is by checking the quality of steam exhausted and compare the quality of the exhaust with the percentage of moisture that was present when the turbine was new and supplied with the same pressure and quality of steam and about the same load. If the exhaust is dryer, that is, shows a less percentage of moisture than formerly, that would indicate less efficient operation, attributable in every probability to worn or eroded blading. However, such a test is only inferential. To make sure that the lower efficiency is due to erosion, the turbine should be sufficiently dismantled for actual inspection of the blading. In most types of small and moderate-sized turbines examination of the condition of the blading can be made by simply removing the steam chest.

QUESTION [Necessity for Air Pump with Surface Condenser]—*Why is it necessary to use an air pump with the surface condenser of a condensing engine?*

H. M.

Answer—Air or other gases held in suspension in the boiler-feed water are finally discharged with the exhaust into the condenser, and in addition, atmospheric air finds its way to the condenser through leaks of the packing and joints of connections with the condenser. Unless the gases thus accumulating in a condenser are removed as fast as received, their presence causes an increasing loss of the vacuum obtained by condensation of the steam.

UE ST]CN [Turbine Exhaust Piping]—*How shall th exhaust piping be connected to a non-condensing turbine?*

Answer—If the turbine is piped directly to the air with a short light pipe, it is necessary only to see that the pipe drains itself, or that the proper drain is installed.

If the turbine is connected to a heavy exhaust header, it is necessary to place a flexible expansion joint in the exhaust connection, and also a valve at the exhaust header, so that the exhaust back pressure can be cut off in case it is necessary to repair the turbine. In medium or larger size machines special care must be used to see that no weight or expansion strains are placed on the turbine by the exhaust pipe. It is better if anchored at the turbine, so expansion moves the other end.

When a valve is placed between the turbine and the exhuast header a relief pop valve should be installed between the turbine and the exhaust valve. Sometimes an exhaust valve may be closed by accident or through a mistake and, also, exhaust valves sometimes gradually work shut themselves while the machine is running. In such a case the turbine would blow up, unless there were an escape for the steam, such as a pop valve. This should be placed where it will not freeze up, and usually is in the engine room.

QUESTION [Lubricating Oil]—*What kind of oil is required for steam turbines?*

Answer—A lubricant free from vegetable or animal oils; from acid, and which should not saponify or foam up when placed in contact with water. Turbine oils are refined from crudes which have either paraffin or asphaltum as a base. Pennsylvania crudes are paraffin base oils and oil from Texas or further south has asphaltum as a base. The former are usually lighter in color than the latter in the refined form suitable for turbines.

QUESTION [Turbine Lubricating Oil Tests]—*What simple tests can be made for determining the quality of turbine oils?*

R. C. W.

Answer—For determining whether an oil is well refined, fill a clean bottle about half full of the oil and heat slowly over an open flame, until vapor appears above the oil surface, and maintain that temperature for 15 minutes. If well refined the oil will darken in color, but will remain perfectly clear and free from sediment after standing for 24 hours; but if the oil has been poorly refined it will turn black, and, after standing, a black deposit will appear.

To test for tarry and other residues place a small quantity of the oil in a test tube along with about 20 times the quantity

of 85 Be. gasoline. Allow the mixture to stand for an hour, and the tar and other insoluble matter present will collect at the bottom of the test tube.

A well-refined oil should show no trace of sulphuric acid. A simple test for acid is made by partly immersing a piece of highly polished copper in the oil for about 24 hours. If a trace of acid is present the immersed surface of the polished copper will be made dull. The simplest test for sulphuric acid is to wash a sample of the oil thoroughly with warm water and test the water with neutral litmus paper. If even a faint reddish tint is shown on the paper the oil should be rejected. The water should first be tested to be sure it is not acid, in itself.

A quick method of detecting the presence of animal oils is to place a few drops of the oil sample in the palm of one hand and rub it briskly with the other hand. The heat produced will cause the animal oil, if present, to emit its characteristic odor. A rough test to determine the presence of animal oils in compounds with mineral oils may be made by heating two ounces of the oil in a small glass or test tube with the same quantity of borax, caustic potash or caustic soda, for about 15 minutes, when the presence of animal oil will be shown by a light yellow deposit.

QUESTION [Turbine Hunts]—*What is the cause of a turbine changing speed rapidly, or so-called "hunting"?*

E. S.

Answer—This is caused usually by friction or lost motion in the valve gears. Friction is the most common cause. With direct connected valve gears the friction is usually found in the steam valve or in the governor parts. With driven gears, such as the hydraulic gear, friction is most likely to occur in the pilot valve which is moved by the governor, in order to let the oil pressure in the operating cylinder and move the valve gear. Too heavy a dashpot will give sufficient friction to cause hunting. There are many other causes of this trouble which cannot be explained in a brief manner.

Lost motion usually produces surges or shaky motions of the valve gear or changes in load. The method of locating these troubles is usually to disconnect part of the valve gear and move it by hand. Lost motion or friction can readily be located this way.

QUESTION [Erection of Turbine]—*We have purchased a second-hand turbine. Please outline the steps to be taken in installing this machine.*

R. D. F.

Answer—(1) Obtain an instruction book from the manufacturer or absolute information about what is required in

installing the particular turbine. (2) Unload the turbo generator and make sure the foundation is clean and ready for the machine. (3) Set the turbine on the foundation with wedges under the bed plate. (4) Line up and level the turbine temporarily. (5) Check up the piping and see if the foundation bolts and all parts will line up with the turbine satisfactorily. (6) Line up the turbine and level it permanently. (7) Grout in the unit. (8) Clean everything and complete assembling. (9) Inspect everything and be sure no bolts or solid object are in the steam or oil passages. (10) Connect the piping.

QUESTION [Starting up a Small Turbine]—*Please outline the procedure to be followed in starting up a small non-condensing turbine.*

J. C. R.

Answer—Open the drains in the wheel casing or cylinder drain water from header above throttle valve and open drain in exhaust header if there is any.

If the turbine is cold and there is steam in the exhaust header, so that the turbine will undergo heating when the valve is open, then open up the exhaust valve gradually so the machine will heat slowly and evenly. If the machine is already warm, then there is no reason to delay opening the exhaust valve.

If there is any water required for packing glands or for oil system see that it is running. Inspect oil level in oil reservoir.

Fill oil or grease cups and oil small part of valve gear, etc., if necessary.

Trip the emergency valve by hand to make sure that it is working.

Turn the turbine over by steam and then shut the throttle valve.

Bring the turbine to speed slowly.

Notice the oil pressure and governor operation, as the machine speeds up, and the main valve closes. See that there is no overspeeding.

Put a light load on the generator and then open up the throttle valve wide.

If it happens that the turbine overspeeds on account of valve leakage it is best to throttle down until the load is applied so as to prevent overspeeding.

After the load is on, the drains should be regulated for running condition. The drains in the steam header should be shut off and the drains in the casing and exhaust header should be either shut off or connected to trap or cracked as the practice of the station requires. It is usual practice to leave the drains to the wheel casing or cylinder partly open, or just barely cracked. The exhaust line ordinarily takes care of its own water without draining, unless there is a pronounced water pocket.

The bearing oil pressure and general mechanism should be looked over carefully to see that everything is O. K. before leaving the machine.

QUESTION [Causes of Turbine Vibration]—*What causes a turbine to vibrate?*

C. S. B.

Answer—Unbalance and misalignment are usually the causes of the few cases of vibration that appear.

Water coming over with the steam will cause multi stage turbines to vibrate but usually does not affect single stage or small machines. Electrical causes which throw a strain on the revolving electrical element cause it to be pulled out of line and produces vibration sometimes.

Other causes of vibration are loosening, warping or failure of parts of the rotor or stator, eccentric or loose coupling, spring shaft, settling of the foundation, uneven expansion, sympathetic vibration, or vibration which is easily set up in the foundation, floor, piping, or other parts connected with the turbine; strains caused by exhaust piping or steam piping, excessive variation of vacuum in some cases, the restriction of oil or cooling water; improper action of the valve gear.

Lack of accurate alignment is the principal cause of turbine troubles and greatest care in the adjustment of steam turbines is required.

This is a complicated subject and cannot be treated briefly with much satisfaction.

The ordinary test for first class operation is to balance a coin on the bearing bolts. If this will stand satisfactorily on edge, the operation is considered good. Small flexible parts, such as brackets, piping, etc , may vibrate considerably although the general operation is good.

Section II

Steam Turbines

SECTION II

Steam Engines

QUESTION [Mechanical efficiency of Engine]—*What is the mechanical efficiency and brake horsepower of an engine, when there is development of* 142 *indicated horsepower and with no load on the engine there is development of* 19.5 *indicated horsepower?* L. M.

Answer—The mechanical efficiency of an engine, or its efficiency considered simply as a machine, is the ratio of the brake horsepower to the indicated horsepower that must be developed for production of the brake power; or stated asa formula,

8430 Stores Power—Book 8-8&159-9 Allen Sept. 16 Galley 15

$$\text{Mechanical Efficiency} = \frac{b.hp.}{i.hp.}$$

The difference between the indicated horsepower and the brake horsepower is the amount of power required to overcome friction. With most steam engines this difference, called the friction horsepower, is practically constant for all loads from 0 to full load, and the brake horsepower developed for any indicated power is assumed to be the indicated horsepower minus the friction horsepower, or horsepower indicated when the engine runs at its regular speed with no load. Hence in the example, b.hp. = 142 − 19.5, or 122.5, and mechanical efficiency $= \frac{b.hp.}{i.hp.} = \frac{142 - 19.5}{142} = 0.86$.

QUESTION [Shortest Economical Cutoff or Plain D Slide Valve]—*Why cannot a D slide-valve engine economically cut earlier than one-half stroke?* C. J. Y.

Answer—In a slide-valve engine cutoff before the end of the stroke is obtained by setting the eccentric more than 90 deg. in advance of the crank, and that hastens all of the valve events. The lead must be corrected by adding lap to the valve. There is then less valve opening and also slower motion of the valve at the points of opening and closing, because those events occur when the eccentric is nearer its dead center, and therefore shortening of the cutoff is accompanied by wiredrawing the steam and there is realization of considerabl less than steam-chest pressure, even at the beginning of the stroke.

Hastening of the release is accompanied by hastening closure of the exhaust. If the release is delayed by adding lap to the exhaust edge of the valve, less exhaust opening is obtained, the greater back pressure is further increased by earlier closure of the exhaust, and the resulting compression may be so excessive as to force the valve from its seat and thus permit leakage of steam from the steam chest directly to the exhaust outlet of the engine. These considerations affecting the action of the plain D slide valve have established a limit to which it may be economically used. It is sometimes employed for cutoff at one-half, but as a rule with a fixed eccentric, not earlier than five-eighths stroke.

QUESTION [Testing Valve Leakage of Single-Valve Engine]—*How can a test be made of valve leakage of a slide-valve or single-valve automatic engine?* C. J. Y.

Answer—A general test of tightness can be made by turning the engine over to such a position that the valve covers the ports of both ends of the cylinder at the same time. Then, upon admitting steam at the throttle valve, leakage will be shown by discharge of steam from open cylinder pet cocks or indicator connections, or by escape of steam from the exhaust pipe. The leakage under running conditions can be approximately determined by blocking the engine wheel and making tests at different points of stroke of the piston. To test valve leakage of a throttling D slide-valve engine at a given point of piston stroke from the crank end of the cylinder, remove the cylinder head and with the piston in the crank end of the cylinder, turn the wheel in the running direction, and block the wheel when the piston has arrived at the desired point; then gradually admit steam through the throttle, and observe whether there is escape of steam from the steam passage of the head end into the cylinder or out of the exhaust pipe.

Piston leakage must be corrected before it is attempted to inspect leakage of the valve when it is in position for admission of steam for a piston stroke from the head end of the cylinder, because the crank end of the cylinder cannot be uncovered to distinguish piston leakage from valve leakage. When the piston packing has been made tight and cylinder head replaced, turn the engine wheel in the running direction until the piston has arrived at the desired point of stroke from the head end of the cylinder. Then with the wheel blocked, open the throttle a little, and steam escaping from the exhaust pipe, or the cylinder pet cock or indicator connection of the crank end, will indicate the valve leakage.

With a single-valve automatic engine, proceed the same as for testing valve leakage of a throttling engine, but with the governor blocked in its average running position, and positions giving other points of cutoff of which it is desired to test valve leakage.

QUESTION [Testing Equality of Piston Clearance]—*How is it determined whether an engine connecting rod is of correct length to give equal clearance in each end of the cylinder?* J. N. E.

Answer—Make a mark on the crosshead and corresponding marks on the guide when the piston is first at one end and then at the other end of the stroke. Then disconnect the connecting rod and move the crosshead so the piston strikes against first one end and then the other end of the cylinder, each time making a mark on the guide opposite to the mark on the crosshead. In each instance the distance between the marks made with the piston in the same end of the cylinder will be the piston clearance in that end when the connecting rod is connected up. These distances can be compared by taking the length of one of them with a pair of dividers and setting it off over the other distance.

QUESTION [Setting Link-Motion Valve Gear]—*What is the method of setting the eccentrics and valve of a link-motion valve gear?* A. I.

Answer—Assuming that the link motion is designed for equal leads, place the link at full gear forward and, with the crank on a dead point, set each eccentric as near as may be with the proper angle of advance, remembering that the centers of the eccentrics should be on opposite sides of and at the same angle (more than 45 degrees) with the crank. Give the valve the proper lead by adjusting the length of the valve rod or forward eccentric rod. Then turn the engine into the other dead point, and, if the valve does not give the proper lead, change the length of the eccentric rod or valve rod to correct half of the error and shift the forward eccentric until the proper lead is obtained. Then place the link at full gear backing, and in the same manner as before obtain the proper lead with the crank on one center and then the other. Again place the link at full gear forward and see if the setting has been disturbed. If it has the valve must be reset by the same method until equal loads are obtained. Usually a single repetition of the adjustments in forward and backing gear will be found sufficient. If the motion is designed for unequal leads, as commonly the case with marine engines, the processes will be the same, only in that the adjustments must be made to the proper lead for each end.

QUESTION [Apparent and Real Cutoff]—*In operation of a steam engine what is meant by "apparent" cutoff and "real" cutoff?* J. D. H.

Answer—Cutoff is the event during the stroke when the steam-admission valve has just closed, preventing further

admission of steam, and the fraction of stroke that is completed at the time cutoff occurs is called the "apparent" cutoff.

During the remainder of the stroke the piston receives the expansive force of the volume of steam in the clearance space plus the volume contained in the space swept by the piston from the beginning of the stroke to the point of cutoff, and the "real" cutoff is the ratio of the sum of these volumes to the volume swept by a complete stroke of the piston plus the volume of the clearance. Hence, to find the real cutoff, add the clearance, expressed as a percentage of the volume swept by a complete stroke of the piston to the apparent cutoff expressed as a percentage, multiply the sum by 100, and divide the product by 100 plus the clearance percentage.

QUESTION [Condensation May Offset Gain from Expansion]—*Why is not cutoff of steam at one-fifth of the stroke, or less, as economical as one-fourth or one-third stroke?*
C. T. S.

Answer—Early cutoff and consequent high rate of expansion increase the amount of cylinder condensation on account of the extreme changes of temperature, and when the cutoff is earlier than about one-fifth, the condensation may offset any gain from increased expansion.

QUESTION [Heat Equivalent to Horsepower-Hour]—*How many heat units are equivalent to one horsepower acting for one hour?*
A. F. M.

Answer—Carefully conducted experiments have shown that taking one British thermal unit as the mean heat required to increase the temperature of 1 lb. of water per deg. F., between 32 and 212 deg. F. gives

1 *mean B.t.u.* = 777.54 *standard foot-pounds*

The quantity is called the mechanical equivalent of one heat unit and, for practical computation, usually is taken as 778 foot-pounds. One horsepower = 33,000 ft.-lb. per minute, or acting for one hour $33{,}000 \times 60 = 1{,}980{,}000$ ft.-lb. Hence $1{,}980{,}000 \div 778 = 2{,}545$ B.t.u. are equivalent to one horsepower acting for one hour.

QUESTION [Use of Poppet Valves With Superheated Steam]—*Why are poppet valves generally used in engines supplied with superheated steam?*
W. A.

Answer—Large valves and valves with rigid sliding surfaces, such as slide valves and Corliss valves, do not work well with highly superheated steam, as the large castings warp so the surfaces do not remain true and the absence of moisture in the steam to act as a seal leads to excessive leakage; and, in

addition, lubrication of the valves becomes difficult with highly superheated steam.

QUESTION [What is the Thermal Efficiency of a Steam Engine]—*What is the thermal efficiency of a steam engine and how is it found?* J. S. P.

Answer—The proportion of the total heat consumption which is converted into work is called the "thermal efficiency," and is found by dividing 2,547 (B.t.u. equivalent to 1 hp.-hr.) by the number of heat units actually consumed per hp.-hr. The quotient is multiplied by 100 to express the thermal efficiency in per cent. The formula is

$$\text{Thermal Efficiency} = \frac{2547}{w(H_1 - q_2)}$$

where

w = pounds of steam as supplied per i.hp.-hr.;
H_1 = Total heat above 32 deg. per pound of steam at the initial conditions prevailing before throttle valve;
q_2 = Heat of liquid above 32 deg. in one pound of water at the temperature of saturated steam at exhaust pressure.

QUESTION [Engine Efficiency]—*What is meant by "the engine efficiency of a steam engine"?* M. R. G.

Answer—The engine efficiency is the ratio obtained by dividing the heat equivalent of the actual work done by the heat available for an ideal engine. The accepted standard for the ideal steam engine is the Rankine cycle. The engine efficiency is obtained by the following equation:

$$\text{Engine Efficiency (referred to i.hp.)} = \frac{2547}{w(H_1 - H_2)}$$

where

w = Pounds of steam per i.hp.-hr.;
H_1 = Total heat above 32 deg. per pound of steam at the initial conditions prevailing before the throttle valve;
H_2 = Total heat above 32 deg. per pound of steam after adiabatic expansion from initial conditions to the final pressure.

H_1 and H_2 can be found from any Total Heat-Entropy diagram. $(H_1 - H_2)$ is the heat available for work.

QUESTION [Leakage of Poppet Valves]—*Why do double heat poppet valves leak when the throttle valve is just "cracked,', although when under full steam the valves apparently do not leak*.

Answer—Such valves are usually ground and tested for leakage at working temperature. When the throttle valve is just "cracked" the temperature is below normal and the valve may expand more than the seat and show leakage.

QUESTION [Equalizing Cutoff of Corliss Engine]—*How is the same point of cutoff obtained for opposite ends of the cylinder of a Corliss engine?* C. R. T.

Answer—Make a mark on the crosshead and corresponding marks on a guide when the piston is at each end of its stroke, and another mark on the guide to indicate where the mark on the crosshead will stand when the piston is at one-quarter stroke from each end of the cylinder. With the wristplate hooked up and the governor blocked to the average running height, place the piston at one-quarter stroke from either end and adjust the length of the governor reach rod to the valve on the same end, so the valve will be just tripped when one-fourth stroke is reached. Then turn the engine over slowly and observe whether cutoff takes place as the piston is brought up to one-fourth stroke from the other end of the cylinder. If cutoff takes place too early, lengthen or shorten the governor reach rod to the valve of the same end so that it will not take place until after one-fourth stroke or if cutoff was found to take place too late, gradually adjust the length of the reach rod so the valve is just tripped when the piston is brought up to one-fourth stroke. If there is a single reach rod from the governor with a link connection between the valves, adjustment should first be made of the rod that has direct connection with the governor. When adjustments are made, test the equality of cutoff when the governor is blocked a little higher and also when blocked a little lower. On account of variation of angular position of the governor connections, it is impossible to make adjustment that will obtain exact equalization for all positions of the governor, but when made for any particular position, equalization for the average load can be made by adjusting the cutoff of one end of the cylinder checked by indicator diagrams.

QUESTION [Proportions of Corliss Engine Cylinders]—*Why are Corliss engines made with cylinders longer in proportion to their diameters than in most other types of cutoff engines?* J. R. F.

Answer—The speed at which an engine can be run with regular Corliss valve gear is limited to the time required for the various parts of the valve gear to adjust themselves and for the dashpot pistons to work properly; and to obtain a proper piston speed with the slow rate of revolution required,

the stroke of piston and length of cylinder are made long in proportion to diameter.

QUESTION [Reducing Speed of Corliss Engine]—*How can the speed of a Corliss engine be reduced from 75 to 60 r.p.m. if the weight on the governor has been set over as far as it will go for reducing the speed?* E. B.

Answer—Ascertain what speed of engine is obtained when the weight for adjusting the speed is set at about the middle of its adjustment. Then, to retain the same speed of the governor with regulation at 60 r.p.m. of the engine, replace the receiving pulley on the governor by one whose diameter is equal to the diameter of the present governor pulley multiplied by 60 and divided by the ascertained r.p.m. of the engine; or replace the governor driving pulley on the engine shaft by one whose diameter will equal that of the present pulley multiplied by the ascertained r.p.m. and divided by 60. The exact change of speed will not be obtained on account of a small error due to neglecting consideration of thickness of the governor belt, but the deviation will be so little from the desired speed that it can be corrected by changing the position of the speed-adjusting weight.

QUESTION [Setting Safety Cams of Corliss Engine]—*What is the method of setting the safety cams of a corliss valve gear?* G. T.

Answer—When the cutoffs have been equalized, drop the governor to its lowest possible position and set the safety cams around to such a position that they will prevent the valves from being picked up, or at least, so each valve will be tripped before its lap is uncovered. Observe, however, that the safety cams are not set so far around that the valve cannot be picked up when the governor is resting on the starting pin or collar. Also place the governor in its highest position and see that the valves are not opened in that position, as that should be the condition to prevent the engine racing in case the load is suddenly removed.

QUESTION [Dashpots of Corliss Engine]—*Why are dashpots used on the valve gear of a Corliss engine?* J. G. S.

Answer—A dashpot plunger is suspended from an arm on the spindle of each steam admission valve to close the valve quickly and quietly after it is disengaged for effecting cutoff. In most forms, while the valve is opening a partial vacuum is obtained in the "vacuum chamber" of the dashpot by the upward movement of the plunger, and when the valve is released the fall of the plunger is accelerated by the pressure of the atmosphere. The "vacuum chamber" plunger has connected to it the piston or plunger of a "cushion chamber," or dashpot proper, in which an air cushion is provided to bring the plungers to rest without shock.

QUESTION [Range of Single-Eccentric Corliss Cutoff]—*With a single-eccentric Corliss engine, why cannot the range of cutoff be obtained through more than one-half stroke by giving the steam valves negative lap and setting the eccentric with negative advance the same as with a double-eccentric Corliss?*
R. G. N.

Answer—Cutoff at later than one-half stroke could be obtained, but setting the eccentric back of the 90-deg. position gives rise to impractical operation of the exhaust valves. An exhaust valve opens and closes for the same position of the wristplate. If the valve has no lap when the wristplate is in its central position, and the eccentric is set behind the 90 deg. position (that is has negative advance), the exhaust valve would not open until after the beginning of the stroke from the opposite end of the cylinder; and as 180 deg. rotation would be required to bring the valve back to the same position, it would not close until the piston had proceeded on the power stroke from that end of the cylinder which contains the valve. If the valve rod is adjusted so the valve has enough negative lap to open earlier, it will remain open all the longer for steam to blow through after the beginning of a stroke from the same end of the cylinder. On the other hand, if lap is given to the valve to prevent steam from blowing through, it will be opened still later in the piston stroke from the other end, and the greater delay of release will further increase the back pressure.

QUESTION [Sizing Bushing for Engine Cylinders]—*We are preparing to rebore and bush a* 26×42 *in. Corliss engine cylinder. The bushing is to be about 1 in. thick and the cylinder will be bored in four steps. How much oversize will it be safe to make the bushing larger than the cylinder diameter to press the bushing in place with a 35-ton jack?* W. D. P.

Answer—No more than a snug fit is necessary when the bushing and cylinder are at the same temperature and appropriate provision is made for preventing the bushing from turning or slipping endwise in the cylinder. Very good jobs have been made by turning off the outside of the bushing about 0.005 of 1 in. greater in diameter than the diameter of the cylinder bore when both are at the same temperature and then inserting the bushing with the cylinder warmed up and the bushing cold. By that method the pressure required for forcing the bushing into the cylinder will depend upon the force required to spring the bushing to the form of the cylinder in compensation of imperfect boring and turning.

QUESTION [Angle of Advance and Lap for Double-Eccentric Engine]—*In order to obtain greater range of cutoff than 0 to* $\frac{1}{2}$ *stroke with a double-eccentric Corliss engine, why is it necessary to set the steam eccentric with a negative angle of advance and for the steam valve to have negative lap?* A. L. S.

Answer—With the Corliss valve gear the valve cannot be tripped to effect cutoff later in the stroke than when the eccentric has carried the wristplate farthest to one side of its central position. When the eccentric is set 90 deg. with the crank, the farthest swing of the wristplate to one side occurs when the piston has reached about one-half stroke, and in order for the maximum wristplate displacement to be delayed so as to permit of cutoff after one-half stroke, it is necessary for the eccentric in the beginning of the stroke to be set back of the 90-deg. position; that is, the eccentric must be set with negative angle of advance. With the eccentric thus set behind the 90-deg. position, if the valve rods were of such length as to bring the valves line on line when the wristplate was in the central position, then with the engine on a center and the wristplate hooked up to the eccentric, the edges of the valves would overlap the ports, and to obtain admission at the beginning of the stroke, it would be necessary to adjust the length of each valve rod so its valve would be brought around to a position ready to open or give the desired amount of lead at the beginning of the stroke. When the valves are thus adjusted, the negative lap, or amount of opening the valves have when the wristplate reaches its central position, is a necessary consequence of adapting the lead to negative angle of advance of the eccentric.

QUESTION Obtaining Compression of Exhaust on Corliss Engine]—*How can compression of exhaust be increased on a Corliss engine?* E. D. C.

Answer—With a double eccentric Corliss engine, advance the exhaust eccentric. As opening and closing of each exhaust valve occurs when the piston is at the same part of its stroke, earlier closure for obtaining more compression will also cause earlier release after cutoff. With a single eccentric for operating all valves, advancing the eccentric hastens all valve events and causes the steam valves to open earlier. If the resulting lead is too much, the time of opening the steam valves must be delayed by adding lap to those valves. This is done by adjusting the length of the steam valve rods so the opening edges of the steam valves will lap over their ports when the wrist plate is in its central position, giving one-thirtysecond to one-sixteenth in. opening for lead when the piston is at the beginning of its stroke.

QUESTION [Click in Engine With a Particular Pressure]—*What may be the cause of a clicking noise that seems to be in the crankshaft of a large Corliss engine whenever the steam pressure drops to 85 lb.? This has taken place for many months, without change, and after careful inspection of the shaft, flywheel crank and crankpin there is no apparent cause for noise from movement of those parts of the engine.* P. F. C.

Answer—The cause of clicking or knock in an engine is frequently difficult to locate on account of transmission of

sound from one part of the engine to another. If the noise occurs only with change to a particular initial pressure, regardless of load, it is most likely due to play of a piston packing ring or of the bull ring. In any event, the cause of a click knock or rattling that is coincidental with certain pressure conditions should be sought for in the cylinder of the engine.

QUESTION [Obtaining Additional Compression on Corliss Engine]—*On a Corliss engine with two eccentrics, should increase of compression be obtained by changing the length of the exhaust-valve rods or by advancing the exhaust eccentric?* W. K. R.

Answer—It is better to obtain additional compression by advancing the exhaust eccentric rather than undertake to preserve pressure of expansion near the end of the stroke where it is of little value. If the cutoff is short, earlier release may be of benefit in preventing the valves from slamming on account of the pressure being carried below the back pressure, while with a heavy load it always is desirable to have release early enough for the cylinder to be cleared of pressure previous to the return stroke.

QUESTION [Selection of Initial Pressure]—*In operating a 150-hp. noncondensing Corliss engine my superintendent claims more steam is used at 80 lb. boiler pressure than there would be for 100 lb. pressure. I believe there is less steam used at 80 lb. pressure. Which is right?* A. E. S.

Answer—Analysis of indicator diagrams taken of the average load probably would reveal which is right. If the load is such that for steam at 100 lb. pressure the required point of cutoff results in expansion to not less than about 10 lb. above the pressure of the exhaust, then it would be more economical to supply the engine with steam at 100 lb. pressure. Otherwise the lower initial pressure would be more economical, for although there would not be as much work of expansion realized from the later cutoff, there would be a higher terminal pressure and the less range of cylinder temperature would result in less loss of heat from the cooling effect of the exhaust than in case of higher initial pressure and a lower terminal pressure.

QUESTION [Hunting Action of Engine Governor]—*What causes hunting action of a flyball governor?* R. D.

Answer—When an alteration of speed begins, the governor does not act immediately, because it can operate only after a change of speed has occurred, and a change of position does not affect the engine speed immediately, on account of both the inertia of the moving parts of the engine and the energy of steam which has passed the control of the governor. By the time the change in engine speed has had full effect on the governor, it is forced beyond the position necessary to bring the speed back to normal. Then for the same reasons the

governor is forced into a position of overcontrol in the opposite direction. Hence hunting action may be the result of over sensitiveness of the governor, or may be caused by friction, or the over-dampening by dashpots of the governor, first causing it to be delayed and then overreaching to a changed position; in slow-speed engines the hunting is aggravated by a wide difference of mean effective pressure in opposite ends of the cylinder.

QUESTION [**Hunting of Shaft-Governed Engine**]—*What makes a shaft-governed engine hunt?* A. H.

Answer—The force of governor springs and effective centrifugal force of the weights are unbalanced and alternate in overcoming each other. The remedy is to give less tension on the springs to decrease sensitiveness, and changing the weight to get the desired speed. Hunting and racing may be caused also by friction of the governor parts or connections. When caused by friction the weights will remain on their inner position until the speed developed is so high as to throw them out with a noise; or when the engine is above speed they will stick where they are until the speed is reduced enough for the springs to draw the weights back again.

QUESTION [**Speed of Governor with Increase of Engine Speed**]—*When an engine is speeded up, does the governor run any faster?* M. F.

Answer—When speeding up is accomplished by a change in size of the driving pulleys or gear wheels, the governor makes the same speed as before, but if speeding up is accomplished by changing the governor weights or springs without changing the governor drive, then there is necessarily the same percentage of increase of the governor speed as of the engine.

QUESTION [**Use of Gear Drives for Throttling Governor**]—*Why are not gear drives used on throttling governors instead of belts?* P. F.

Answer—Many of the early designs of throttle-governed engines were provided with governors driven by spur wheels and connected to the throttle valves by reach rods, or when placed near the engine cylinder the governor was driven from the engine shaft by bevel gears and a lay shaft. But gear drives are expensive to construct, more or less noisy, dangerous and easily disabled by the gear teeth becoming stripped and for most situations belt drives are preferable on account of their smooth and noiseless running, ease with which pulley diameters can be adapted to the desired speeds, and safety against stripping of necessary governor gearing.

QUESTION [**Increasing Size of Governor Driving Pulley**]—*If a reduction is made in the diameter of a pulley on the main shaft of an engine for driving a flyball governor, what effect will it have on the speed?* W. T.

Answer—Compared with the speed of the engine, the governor will be driven at relatively slower speed and the engine shaft will have to attain a higher speed before the speed of the governor is sufficient to check the supply of steam. Consequently, reducing the size of governor driving pulley on the mainshaft, or increasing the size of the governor receiving pulley, would cause the governor to regulate the engine at a higher speed.

QUESTION [Operation of Turbine Governor]—*With a hydraulic turbine governor, will the governor remain in the same position as long as speed of the turbine is the same, or do the governor weights stand at different positions for different loads on the turbine?* W. G. B.

Answer—When a turbine is governed by the indirect, or what is commonly called the "relay" method, the centrifugal force of the governor is needed only to "give the signal," which sets in motion an auxiliary mechanism by which the valves are moved by gearing connected to the main shaft or by steam or by hydraulic pressure. The signal is given for more or for less speed, according to requirements and continues until the correct speed is satisfied. There is the same speed and therefore the same standing of the governor weights when the speed is satisfied with one load as with another, within the range of variation required for operation of the pilot valve.

QUESTION [Knock in Low-Pressure Cylinder with Light Load]—*What would cause a knock to set up in the low-pressure cylinder of a compound condensing engine whenever the load is reduced and the vacuum goes from* 26 *in. to about* 28 *in.* R. H.

Answer—If the knock with the lighter load is heard like a slapping noise, as though the exhaust valves are driven from their seats, it may be attributed to expanding the steam below the pressure of the exhaust. But if the knock is in the nature of a pound, such as might be due to slack connecting-rod brasses, it probably would be due to insufficient compression of the exhaust for good cushioning effect. To obtain compression and cushioning on the exhaust with the lighter load and higher vacuum, the lower pressure of the exhaust will require the exhaust valves to be set so they will be closed earlier in the stroke.

QUESTION [Why Equal Cutoff Required Unequal Valve Laps]—*To obtain equal cutoff with an ordinary D slide-valve engine, why is it necessary to have less outside valve lap for the crank end than for the head end of the cylinder?* C. R.

Answer—On account of the angularity of the connecting-rod, the first half of a stroke of the piston from the crank end of the cylinder is accompanied by more than one-quarter of a complete revolution of the crankshaft, and the first half of

a stroke from the head end is accompanied by less than one-quarter of a revolution. Hence, for any fraction of stroke from the crank end the eccentric is carried through more degrees of rotation than for an equal fraction of stroke from the head end, and if the valve had equal laps, cutoff would occur earlier in the stroke from the crank end than from the head end, and in order to delay closure of the steam port on the crank end to the same fraction of stroke as cutoff occurs for the head end, it is necessary to have less lap of valve for the crank end.

QUESTION [Effect of Increasing Weight of Governor Balls]—*Would increasing the weight of the flyballs of an ordinary Corliss engine governor cause the engine to be regulated at a higher or a lower speed?* I. W. C.

Answer—In operation of the pendulum type of governor ordinarily employed on Corliss engines, the arms from which the flyballs are suspended do not attain as great height as would be assumed by a simple revolving pendulum, because part of the centrifugal effort is employed in supporting a central weight, or in overcoming the elasticity of a spring, in addition to sustaining the weight of the balls and arms. Increasing the central weight, or, as well known, adding load to the governor, acts as a greater restraint that results in increase of speed. The cutoff becomes longer because the balls are in lower position for the same speed, and increase of speed continues until the centrifugal effect becomes sufficient to carry the additional burden laid on the governor. Evidently, the same result of speeding up the engine would result from making the governor balls lighter, but the reverse result is obtained either by taking load off the governor or by increasing the weight of the governor balls; that is to say, increasing the weight of the governor balls will result in slower speed of the engine.

QUESTION [Falling and Rising Exhaust Line]—*What is indicated by falling and rising of the exhaust line of an indicator diagram?* M. S.

Answer—A falling and rising exhaust line may be due to contracted exhaust ports or pipes that do not allow the steam to pass out as rapidly as it is displaced by the piston at its greatest velocity, thus creating greater back pressure toward the middle of the stroke; or a rise of pressure may be due to leakage of the piston or steam valve.

QUESTION [Selecting Scale of Indicator Spring]—*What determines the best scale of indicator spring for taking a diagram?* L. L. S.

Answer—For most purposes it is desirable to use the lowest scale of spring and secure the highest diagram obtainable within the limits of vertical motion of the pencil and good operation of the instrument. Standard patterns of indicators

provide for a pencil rise of 2 in. to $2\frac{1}{2}$ in. above the atmosphere line, and the lowest scale of spring to be employed is found by dividing the initial pressure by the permissible rise in inches. If there is a waviness in the admission or expansion lines of the diagrams due to inertia of moving parts of the instrument, smoother, although lower, diagrams are obtainable by use of a higher scale of spring in the indicator.

QUESTION [Scale of Indicator Diagram for Reduced Size of Piston]—*When indicators are quoted as having one-half or one-fourth size pistons, are the fractions intended to signify inches of diameter or relative areas; and when the smaller pistons are used, how is the scale value of the diagrams determined when the indicator is employed with a standard scale of spring?* G. L.

Answer—Some indicators are made to receive pistons of smaller area than the regular or standard area of piston, by elongating the piston rod enough for a smaller piston to reach down and work in a part of the cylinder where the bore is one-half of one-fourth of the cross-sectional area of the standard piston. By reason of resisting pressure exerted on only one-half or one-fourth of the standard area, the effective scale of the spring thereby becomes twice or four times as great. Thus the scale of a diagram made with a spring whose scale is 40 lb., when used with the standard piston would be 2×40, or 80 lb. per sq. in. per inch rise of the pencil when used with a one-half size piston; and a 60-lb. spring used with a one-fourth piston would produce a diagram whose pressure scale per inch rise of the pencil would be 4×60, or 240 lb. per square inch.

QUESTION [Striking Atmospheric Pressure Line of Diagram]—*Should the atmospheric pressure line of an indicator diagram always be drawn on the card and be made before unhooking the cord from the reducing motion, and is it not equally well to strike the line afterward while drawing out the cord by hand?* T. A. A.

Answer—The atmosphere line is the base line of the diagram, and while not necessary to be taken into account in measuring the diagram, excepting as a convenience, it is necessary for complete interpretation of the diagrams. In all cases the atmosphere line should be struck on the card immediately after diagrams are taken so it will be located as nearly as possible under the same conditions as those under which the diagrams were made. Subsequent drawing of the atmosphere line is likely to give it a false position, and when the cord is drawn out by hand it becomes necessary to "square up" the card to determine the length of a diagram.

QUESTION [Advantages of Inertia Governors]—*What are the advantages of inertia governors over simple centrifugal shaft governors?* C. B.

Answer—Inertia governors act quicker in adapting the point of cutoff to a change of load and are capable of a design in which the inertia and centrifugal effects are so combined that, for a change of load, the speed of the engine is held closer to the normal speed than when only centrifugal effect is depended upon.

QUESTION [Obtaining Increase of Speed with Shaft Governor]—*How would the speed of a shaft-governor engine be increased?* G. E.

Answer—The speed will be increased by increasing the resistance that the governor springs exert in opposing movement of the weights from centrifugal force, or by decreasing the weights, or by moving the weights nearer to the center of rotation. In adjusting most shaft governors, when the proper sensitiveness has been reached the change should be made in the weights alone.

QUESTION [Indicator Diagrams from Compound Engines]—*For the purpose of considering the valve setting of a compound engine, is it admissible to rely upon indicator diagrams of one cylinder taken at a different time from diagrams taken from the other cylinder?* H. R. J.

Answer—The diagrams should be taken simultaneously, or at least so nearly at the same time as to insure the same conditions of load, engine speed, initial pressure, points of relative cutting-off and same back pressure. Otherwise the actual distribution of steam is largely conjectural, for the diagrams are not truthfully representative of relative performances of the cylinders.

QUESTION [Computing Indicator Diagrams Separately]—*In computing the indicated horsepower shown by indicator diagrams from a double acting engine, should the power be figured separately for the two ends of the cylinder, or may the net piston areas and mean effective pressures for the two ends be averaged?* R. B.

Answer—For close computations, the power developed in each end of the cylinder should be computed separately as two single-acting engines, taking the sum as the total power indicated. When the values of net piston area and mean effective pressures are averaged, the calculation generally will result in a close enough approximation for most practical purposes, provided the mean effective pressures are nearly the same.

QUESTION [Variation in Length of Indicator Diagrams]—*What causes difference in length of indicator diagrams taken with the same indicator and reducing motion from opposite ends of an engine cylinder?* J. B. S.

Answer—Different lengths of diagrams may result from use of indicator cords of different diameter or a different winding

or building up of cords on the reducing motion or sheave of the paper drum. There will also be a shortening of the diagrams due to stretching of the cord or insufficient drum-spring tension to overcome the drum and pencil friction.

QUESTION [Lower Compression Due to Higher Vacuum]—*Why do indicator diagrams from the low-pressure cylinder of our compound condensing engine show less compression of the exhaust for 28½ than for 26 in. condenser vacuum, without change of the engine valve setting?* A. H.

Answer—The absolute pressure represented by 28½ in. vacuum would be $30-28\frac{1}{2} = 1\frac{1}{2}$ in. mercury-column pressure, and for 26 in. vacuum would be $30-26 = 4$ in. mercury-column pressure. Hence, with the exhaust closing at the same fraction of stroke, compression of the exhaust with 28½ in. vacuum would be only $1\frac{1}{2} \div 4$ or $\frac{3}{8}$ as high in absolute pressure as for 26 in. vacuum.

QUESTION [Negative Part of Indicator Diagram]—*What is meant by the negative part of an indicator diagram?* W. R.

Answer—A negative portion of a diagram is any part in which the pressure during the forward stroke of the piston is less than the counterpressure indicated for the same part of the diagram during the return stroke of the piston. When an engine is underloaded, the fraction of stroke at cutoff may be so small that expansion during part of the forward stroke goes below the back pressure during the return of the piston over the same part of the stroke, forming a loop like *BCDEB* shown in the sketch; or the compression line *FG* may rise above the admission line, forming a loop like *GHJ*. The areas contained within such loops are said to be negative because they represent work opposed to the useful work, and in determining

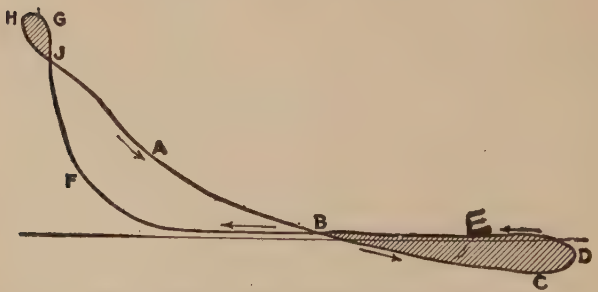


DIAGRAM WITH NEGATIVE AREAS SHADED

the m.e.p., the whole length of the diagram is taken and the negative areas are to be deducted from the area of the rest of the diagram.

QUESTION [Testing Indicator Reducing Wheel]—*How can the accuracy of an indicator reducing wheel be tested?*

E. T. B.

Answer—Set the indicator on the engine or any stationary support, and fix a rod or board having a straight edge that will be followed by the cord when drawn out a distance equal to the stroke of the engine, and lay off the length of stroke along the straight edge and divide it into four or more equal parts, and drive small and equal-sized wire nails at the points along the straight edge that represent the ends of the stroke, and at each of the subdividing points. Then, with a spring in the indicator and a blank card on the paper drum, strike a long atmospheric line by drawing out the cord by hand and light pressure on the pencil. Having the cord of proper length so it will not cause the drum to strike its stop when the hook is on either of the nails that represents the end of the stroke, place the hook on one nail after the other, each time lightly pressing the pencil on the card, and making a short mark across the atmospheric line. If the reducing motion is correct, these marks will be equally spaced.

QUESTION [Piston Ring for 12-in. Steam Cylinder]—*What should be the size of outside diameter for turning a cast-iron piston ring for a steam-engine cylinder* 12 *in. in diameter?*

E. M.

Answer—For a finished ring of about $\frac{5}{8}$ in. thickness and made of a good quality of close-grained cast iron, the outside should be turned off to $12\frac{1}{4}$ in. diameter and have $\frac{7}{8}$ in. cut out of the circumference, or simply cut square across with the ends finished to form a half lap-joint $\frac{7}{8}$ in. long and width equal to the radial thickness of the ring, so the plane of the joint will be parallel with the flat sides of the ring.

QUESTION [Loosening Piston Rod from Crosshead]—*How can the tapered end of an engine piston rod be started out of the crosshead?*

R. G.

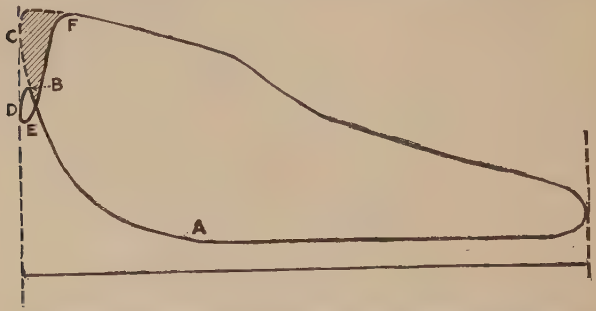
Answer—Place the engine on the head-end center, remove the cotter and replace it by a cotter or piece of steel about one-eighth inch less in width than the clear space through the cotter way. Then detach the eccentric rod and, with the steam valve closed on the head end and open on the crank end, admit steam to the crank end of the cylinder. If the piston rod is not immediately released from the crosshead, clean the parts thoroughly and make another trial after leaving the joint moistened with kerosene for a few hours.

QUESTION [Engine Piston Over-riding Indicator Hole]—*If the piston of an engine travels over the hole in the cylinder for the indicator piping, how does it affect the diagram?*

W. L. R.

Answer—When the piston over-rides the hole in the end of the cylinder for the indicator connection, communication is temporarily cut off between the interior of the engine cylinder

and the indicator cylinder, and if there were no leakage past either the indicator piston or the engine piston, the pressure of steam would remain constant and a short straight line would be traced on the diagram from the point in the return stroke at which the engine piston covered the opening to the end of the stroke, and the line would be retraced until the piston uncovered the hole. But there is always free leakage past the indicator piston and generally past the engine piston. Hence, there usually is a small drop of pressure indicated until the engine piston has again uncovered the hole, when the indicator pencil immediately rises to the point correctly corresponding with the pressure in the engine cylinder, the drop and rise forming a loop at the end of the diagram.



A diagram with such a loop is shown by the full lines of the illustration. The indication of increase in compression of the exhaust from A toward the end of the stroke is arrested at B by the engine piston covering the indicator connection and falls to D at the end of the exhaust stroke, whereas the engine cylinder pressure continues to rise and should be indicated by continuing the compression line from B to C. The indicator pencil continues to fall from D to E in the beginning of the admission stroke and at E the hole is again uncovered and the pencil rises to F in response to pressure within the cylinder, but without embracing the shaded area BCF, which should have been included in the diagram.

QUESTION [Designing Angle of Advance of Eccentric]—*Why is the angular advance of an engine eccentric referred to the position of 90 degrees with the crank in place of designating the whole number of degrees in advance of the crank?*
W. R. T.

Answer—The 90-degree position is taken as the basis for consideration, because in the simplest form of valve, without lap or lead, that position is necessary for admission of steam to one end or the other end of the cylinder, and departures from that form of valve or its operation are naturally referred to the 90-degree position of the eccentric, and "angle of

advance" is commonly so understood. When otherwise intended, the angular advance should be specified as degrees "ahead of the crank."

QUESTION [Effect of Reduction of Diameter of Eccentric]—*What would be the effect in operation of an engine eccentric if $\frac{1}{4}$ in. was turned or worn off the diameter of the eccentric?* H. R.

Answer—A quarter-inch reduction of diameter of the eccentric, without equal reduction of the bore of the eccentric strap, would give rise to $\frac{1}{8}$ in. lost motion at each end of the stroke of the eccentric rod, with a corresponding reduction in the length of valve travel. But if the eccentric straps are bored out to as snug a fit as they first had with the eccentric, and the length of the eccentric rod, measured from the center of the eccentric to the pin in the end of the rod, is made the same as it was before reduction of the diameter of the eccentric, then there will be no change from the original travel of the eccentric-rod pin and the original length of rocker arms and valve rod will give the original valve travel.

QUESTION [Apparent Inequality of Cutoff]—*What would cause a difference in the relative length of cutoff shown by indicator diagrams taken from a single-eccentric Corliss engine with no change of adjustments except setting the eccentric ahead of its former position?* V. W. P.

Answer—For the same load, moving the eccentric would slightly change the relative points of cutoff by introducing a change in the angularity of the valve gear; and diagrams taken from opposite ends of the cylinder may appear to indicate greater inequality of cutoff on account of greater difference of governor positions when the diagrams are made, due to the hunting action of the governor. The true relative cutoff of opposite ends of the cylinder should be determined by indicating them simultaneously as nearly as possible, with the governor held in a fixed position, long enough to strike the diagrams.

QUESTION [Equal Loads and Inequality of Cutoff]—*In a slide-valve engine, where the valve has equal laps at each end, will the leads and cutoffs be equal?* R. C.

Answer—With equal laps there will be equal leads, but on account of the angularity of the connecting-rod, more angular rotation of the shaft and eccentirc is accomplished when the piston traverses the crank-end half than when it traverses the head-end half of the stroke, and consequently the valve will be closed with cutoff accomplished earlier for the stroke from the crank end than from the head end. By moving the valve on its stem toward the head end, the cutoff of that end will be made earlier and cutoff in the crank end will be made later, but the lead of the head end will thereby be reduced and lead on the crank end increased.

QUESTION [**Equalizing Cutoff of Single-Valve Automatic Engine**]—*How can a single-valve automatic engine have the valve set to obtain the same point of cutoff in both ends of the cylinder?* H. M.

Answer—The cutoff can be equalized for a given load by correctly adjusting the length of the valve rod, but with most valve gears there is some variation of the relative cutoff for different loads, and it is best to obtain equality of cutoff when guided by indicator diagrams taken when the engine carries the average load. In the absence of an indicator, make a mark on the crosshead and corresponding marks on a guide to register when the piston is at $\frac{1}{4}$ stroke from each end of the cylinder. Then, with the piston in one end, turn the wheel forward until the mark on the crosshead comes opposite to the mark that determines completion of $\frac{1}{4}$ stroke from the same end of the cylinder, and with the crosshead held at at that point, block the governor to a position that will cause the valve to be just closed. Next, turn the wheel forward past the dead center of the other end of the cylinder until the mark on the crosshead comes opposite to the mark on the guide corresponding with $\frac{1}{4}$ stroke from that end, and observe whether the valve just covers the steam port to the same end of the cylinder. If the valve has been carried to a different position, adjust half of the difference by lengthening or shortening the valve rod. Then reblock the governor so that the valve will be just closed and thus readjust the length of valve stem and governor blocking with the piston at $\frac{1}{4}$ stroke alternately from opposite ends of the cylinder, until the same blocking will just obtain closure of the steam valve for $\frac{1}{4}$ stroke from either end of the cylinder.

QUESTION [**More Wiredrawing of Steam With Earlier Cutoff**]—*When earlier cutoff is obtained with a slide-valve engine by adding lap to the valve and advancing the eccentric, why is there more drop of the pressure during admission and more gradual cutoff, than with less advance of the eccentric?* W. R. C.

Answer—Assuming that the engine shaft rotates at a uniform speed, the valve is moved at the highest velocity when the eccentric is at right angles with the eccentric rod, or neglecting angularity of the eccentric rod, the quickest movement of the valve in opening or closing and greatest opening would be obtained with no lap and no angle of advance. By advancing the eccentric and adding lap, the valve opens after it has passed the point in its travel where it has greatest velocity and is slowing down more and more as it nears the end of its stroke, and complete cutoff occurs when the valve has attained only the same velocity in its return stroke. Hence, the earlier the cutoff obtained by advancing the eccentric the slower and less the amount of opening, and the slower the movement of the valve as it closes, thus giving rise

to wiredrawing of the steam during admission and more gradual cutoff.

QUESTION [Advantages of Compound Engines]—*What are the advantages of compound engines over simple engines?* J. R. M.

Answer—Saturated steam cannot be expanded more than four or five times in one cylinder without so great a range of temperature that the losses from cylinder condensation overcome the advantages of using the steam expansively. If steam of high pressure and temperature is introduced into a small cylinder, and then carried at lower pressure into the main cylinder and there expanded to the desired terminal pressure, the temperature range will not be excessive in either cylinder and, at the same time, the expansive force of the steam is fully realized. In addition the hotter steam thus comes in contact with the smaller areas of the high-pressure cylinders and there is less surface for condensation. Another advantage of compounding is that a simple engine with early cutoff and large ratio between the initial and terminal pressures is subjected to great fluctuations of stresses, requiring a stronger frame and running parts and heavier flywheel, while in a compound engine tnere may be the same ratio of expansion with much less fluctuation of energy of expansion causing less fluctuation of energy in the cylinders; and where the cross-compound type of engine is employed, with the cranks set 90 deg. apart, the turning effort is more uniform and the engine runs more steadily than with a single-crank engine.

QUESTION [Turbine a One-Speed Machine]—*Can a steam turbine be constructed that will develop a normal rated power at different speeds?* A. L. M.

Answer—The turbine is a one-speed machine. For good economy the speed of the rotating buckets must bear a definite relation to the velocity of the steam as determined by the design. Otherwise there will be wasteful impact of the steam against the vanes and the speed of the turbine cannot be changed from that of best economy without serious loss of efficiency.

QUESTION [Anchorage for Base Elbow of Turbine Exhaust]—*A steam turbine running under* 28 *in. vacuum has a* 30-*in. diameter exhaust opening to which a copper expansion joint is bolted, and there is a base elbow bolted to the expansion joint, which is anchored to a concrete pier. What should be the weight of the pier?* R. J. D.

Answer—A vacuum of 28 in. $= 28 \times 0.491 = 13.75$ lb. pressure per square inch less than the pressure of the atmosphere. The area of opening of the elbow is $30 \times 30 \times 0.7854 = 706.86$ sq. in. Assuming perfect flexibility of the expansion joint and elbow connections and neglecting the weight of

the elbow, the excess external pressure tending to lift the elbow and weight of concrete required for its anchorage would be $13.75 \times 706.86 = 9{,}719$ lb. The actual weight of concrete required would be reduced by the weight of the elbow and any upward thrust exerted by the expansion joint.

QUESTION [**Selection of Most Economical Degree of Superheat**]—*In designing a new turbine station, how is the most economical degree of superheat arrived at?* P. A. Y.

Answer—There generally will be saving of 2 to 4 per cent in the cost of fuel for each 100 degrees of superheat of steam at 150 lb. gage pressure. The use of a high degree of superheat is a matter of first cost and cost of maintenance against the price of fuel, depending also on the type of turbine, the load factor, the size of the units and the nature of the service as regards severe and frequent variations in the load; also having in mind the practical operation of suitable superheaters, steam piping, valves, pumps and auxiliary machinery.

QUESTION [**Engine Piston Leakage**]—*Will a piston leak steam when traveling even though it may seem to be right when blocked and tried; and could there be such a thing as leakage past the rings when changing the direction of travel at the ends of the stroke due to side play of the rings.* H. L. K.

Answer—It is practically impossible to prevent all piston leakage. The wear of the cylinder is not uniform for the full length of the stroke, and testing the tightness of the piston at one point does not demonstrate its tightness at other points along the stroke. Leakage may be introduced by side slippage of the rings from reversal of the stroke, but this, like leakage shown by blocking the piston at any point, may be only momentary, while the piston is traveling over that particular point, and it should not be taken for granted that the degree of tightness or rate of leakage found with the piston blocked is the rate of leakage during regular operation. The leakage when running will be considerably less than the highest rate of leakage observed with the piston blocked at a point that shows the most leakage.

QUESTION [**Travel of Cylinder Over Piston**]—*I claim that at times the cylinder of a locomotive travels over the piston, while B and C claim it would be impossible. Which is right?* F. J. S.

Answer—Motion of a body is change of its position with reference to the position of a point, a line, or another body. Regarding a railroad track as a stationary object to which motion of the parts of a locomotive are referred, when the locomotive is going forward, the cylinder is carried forward at the speed of the locomotive, and during a stroke of the piston from the head end of the cylinder, it is not moving forward as fast as the cylinder and the cylinder may be said to pass over the piston in the same way that a boy might

make a running jump and pass over the head of another who is walking slowly in the same direction.

QUESTION [**Piston Clearance and Clearance Volume**]—*What is meant by clearance of an engine?* W. R. G.

Answer—The term clearance is used in two senses. It may refer to the distance the piston could be separately moved beyond its position at the end of the stroke before it would come in contact with the head of the cylinder, or the term may be used with reference to the volume of space in the end of the cylinder and connected passages, from the piston to the valves, when the piston is at the end of its stroke. To avoid confusion the former is called "piston clearance" and is always a lineal measurement, and the latter is termed simply "clearance" and the amount generally is quoted as a percentage of the volume swept through by one stroke of the piston.

QUESTION [**Placing New Piston Rod in Engine**]—*In putting in a new piston rod to replace an old one on a simple engine with V-shaped guides, what is the proper method of centering the rod?* J. K.

Answer—In providing a new piston rod, the centering of the rod with the stuffing-box is likely to become misplaced. First of all, extend a line through the center of the cylinder past the guides and determine whether the faces of the guides are parallel with the cylinder center line, for that purpose measuring from the cylinder center line to a short round shaft or mandrel, placed in the V of the guide, and of suitable diameter to touch the sides of the V at about the middle of the wearing surfaces. If not parallel, the guides need to be made so by adjustment or as a repair or alteration of the engine. When the guides are parallel to the engine center line, place the new piston rod in the piston and with the piston in the crank end of the cylinder and the end of the rod parallel with the guides, adjust the packing rings so the rod will be in the center of the piston-rod stuffing-box. Then put the crosshead in place with necessary adjustment of the crosshead slippers to make the piston rod parallel with the guides when the rod is connected to the crosshead. If all adjustments have been properly made, the rod will travel centrally through the stuffing-box. If it does not, the vertical and horizontal adjustment of the crosshead should be made so there will be no movement of the stuffing-box gland when the engine is running, otherwise the rod will become scored and it will be difficult to make the packing hold tight.

QUESTION [**Average Tangential Pressure on Crankpin**]—*What is the average tangential pressure on the crankpin of a* 14-*in.* × 20-*in. engine where the m.e.p. is* 40 *lb.?*
C. B. H.

Answer—Neglecting the reduction of piston displacement from presence of the piston rod, the average total pressure acting on the piston would be $14 \times 14 \times 0.7854 \times 40 = 6157.6$ lb., and with 20-in. stroke the work performed per revolution would be $6157.6 \times 20 \times 2 = 246{,}304$ inch-pounds. Without allowance for loss from friction the same amount of work would be transmitted per revolution to the crankpin; and as in one revolution the length of path described by the crankpin center would be $20 \times 3.1416 = 62.832$ in., the average tangential pressure would be $246{,}304 \div 62.832 = 3{,}920$ lb.

QUESTION [Enlargement of Crankshaft Diameter]—*Why are crankshafts of engines made larger at the center than at the ends?* J. W. H.

Answer—Crankshafts have enlarged diameters at the middle of their length to afford greater strength and stiffness in resistance of bending stresses due to the weight of the flywheel, and tension of belt or thrust of driving gears, and also to compensate for the reduction of strength from removal of material for a keyway.

QUESTION [Eccentric Ahead or Behind Crank]—*When is the crank of an engine said to follow the eccentric or the eccentric to follow the crank?* H. C. R.

Answer—The angle that an eccentric makes with the crank is indicated by the position of the "high side" of the eccentric with reference to the crank. The crank is said to follow the eccentric or the eccentric to be ahead of the crank, when the eccentric is less than 180 deg. ahead, measuring in the direction of rotation of the shaft. The eccentric is said to follow the crank or the crank to be ahead of the eccentric when the crank is less than 180 deg. ahead of the eccentric, measuring in the direction of rotation.

QUESTION [Advantages of Eccentric]—*When and why is an eccentric used in preference to a crank?* J. D.

Answer—An eccentric is, in fact, a crankpin that is so large as to envelop the shaft. For imparting a smooth reciprocating motion, the eccentric form of a crank is preferable, because the necessary bearing surface takes up less length along the shaft, the crank motion is obtained without cutting away or reducing the strength of the shaft, and may be made adjustable lengthwise on the shaft and also in angular position with respect to the main crank or other parts carried on the shaft.

QUESTION [Angle of Advance of Eccentric]—*What is the angle of advance of the eccentric of an engine?* B. R.

Answer—It is the angle that the eccentric is placed ahead of a line which is at right angle to the crank. Where the valve is moved by direct connections from the eccentric, the angle of advance is the number of degrees the eccentric is

set more than 90 degrees ahead of the crank. Where a reversing rocker is used, the angle of advance becomes the number of degrees the eccentric is set less than 90 degrees behind the crank.

QUESTION [Testing Parallelism of Engine Guides]—*How can a test be made to determine whether the guides of a horizontal engine are parallel with the center of bore of the cylinder without removing the piston, piston rod or crosshead?*
D. A. S.

Answer—The center line of an engine cylinder usually is considered to be the line passing through the centers of the counterbores. However, if it is desired to make comparison of the guides with the center of the bore, it would be well to assume the center line of the bore to be a line through the centers of the bore at each end of the stroke of the piston. To establish such a center line without stripping the engine of the piston, piston rod or crosshead, remove the cylinder head and place the crank on dead-center away from the cylinder, to bring the piston at the crank end of the stroke. Wedge a wooden rod across the cylinder bore near the piston for holding one end of a light but strong line at the center of the bore, and secure the other end of the line to a stationary object at a distance of about six diameters of the bore beyond the head end. Draw the line as tight as possible and adjust the ends of the line so it will be central with the bore vertically and horizontally at both ends of the stroke of the piston. Then tightly stretch a second line of the same kind from the top of the cylinder, vertically over the first line and parallel to it, by careful measurements and plumbing from one line to the other. Stretch a third line parallel to the first and second lines, over or to one side of the cylinder the full length of the first lines, and beyond the guides, for a reference line. Carefully test the parallelism of the reference line with both the first and second lines, using a distance piece made of a light wooden rod with ordinary pins in each end. If in any case careful sighting along a line reveals sag, stretch another line a short distance above the sagging line by careful plumbing and, by replumbing, locate a supporting point for the line at about the middle of its span. Fasten to the crosshead a pointer terminating in a V opening that all but touches the reference line and after rechecking all adjustments remove the line and blocked rod from the cylinder. Then to test parallelism of the guides slowly turn the engine over, preferably backward, and observe whether the crosshead carries the pointer uniformly at the same distance from the reference line for the full length of a stroke.

QUESTION [Front of Engine]—*What is the front of a horizontal engine?*
A. C. U.

Answer—The front is the face or part opposite to the cylinder end of the engine. The cylinder end is commonly

called the "back" end. These terms are not self-explanatory, and "cylinder end" and "crank end" are better. The only excuse for "front" and "back" end would appear to be that they are consistent with calling the stroke of piston from the outboard end of the cylinder the "forward" stroke, in agreement with the usual direction of the power stroke of a single-acting engine, and for the stroke to be "forward" it would have to be toward the "front."

QUESTION [Running Clockwise and Counterclockwise]—*When is an engine said to be running "clockwise" or "counterclockwise"?* R. E.

Answer—A horizontal engine is said to be "running over" when the crank rises at the beginning of the stroke that occurs while the piston is moving toward the crankshaft or when the top of the flywheel turns away from the cylinder, and the term "running under" is applied to rotation in the opposite direction. Stationary engines usually are designed to "run over," so the pressure between the crosshead and the crosshead guide, due to the angularity of the connecting-rod, comes on the lower side of the crosshead only, and also the belt, which usually leads away from the engine, will have the driving pull on the lower side. Hence the direction for running over is sometimes referred to as "running forward." The term "running clockwise" usually is intended to mean "running over," or forward, and in the same direction as the hands of a clock to an observer viewing an engine with the shaft to his right hand and the cylinder to his left. The terms clockwise and counter-clockwise applied to an engine are often confusing, as the direction will be clockwise to a person standing on one side and counterclockwise to one standing on the other side, and it is best to confine the designations of directions of rotation to the terms "running over" and "running under."

QUESTION [Suspended Engine Foundation]—*How can a suitable foundation be provided for an engine placed on the second floor of a steel-frame building so the engine bed may be the usual height above the main floor-line?* F. K.

Answer—The foundation recommended by the engine builder is for the purpose of absorbing vibrations as well as for supporting the engine, and if the building framing is strong enough to support the engine and foundation, a mass of concrete equal to that recommended by the engine builder may be suspended from the floor framing, with the upper surface of the foundation carried up to the desired level for receiving the engine.

QUESTION [Running Over or Under]—*Is more power derivable from an engine running over than running under, and if so, why?* J. B.

Answer—When running over, pressure from the crosshead is on the lower guide and '- due to reaction of the connecting-rod and weight of the crosshead, and much weight of the piston rod and connecting rod as is supported by the crosshead. When running under, the pressure from reaction of the connecting-rod comes on the upper guide, but it is reduced by the weight of the crosshead and parts supported by the crosshead. Therefore in running under the guide friction is less and if anything the actual or brake power developed for the same mean effective pressure the piston would be greater. But under ordinary conditions the friction of guides is such a small percentage of engine friction that the difference of friction running over or under would be imperceptible. The principal advantage of running over is that with the pressure from the crosshead downward there is better opportunity for lubrication of the guide that receives the principal wear, better opportunity for adjusting and maintaining the alignment of the crosshead and less tendency of the crosshead to pound at the ends of the stroke.

QUESTION [**Angularity of Connecting-Rod**]—*What is meant by angularity of the connecting-rod of a steam engine?*

H. G. S.

Answer—The term refers to the angular position of the rod at any time when the center line of the rod, passing through the center of the crank and crosshead pins, is at an angle with the direction of stroke of the piston.

QUESTION [**Superheated Steam for Reciprocating Engines**]—*What benefit is derived from supplying superheated steam to a reciprocating engine and what degree of superheating is advantageous?*

T. B. S.

Answer—The main benefit derived from supplying a reciprocating engine with superheated steam is to counteract cylinder condensation and assure the presence of dry steam at the time cutoff occurs; and there is additional gain in engine efficiency from superheating the steam to such a high degree that the steam will be superheated during the exhaust.

The degree of superheating necessary for insuring dry steam at cutoff depends on the conditions. Ordinarily, this will be obtained by having the steam, as delivered at the engine, superheated about 75 deg. above the temperature of dry saturated steam of the given pressure; and to avoid any cylinder condensation from admission to exhaust usually requires about 150 deg. of initial superheating. The over-all economy of superheating the steam depends on the plant-operating conditions.

QUESTION [**Turning Exhaust into Steel Stack**]—*We are annoyed by exhaust steam from our engine being blown against the side of our factory building. Could not the exhaust be discharged into the* 40-*ft. iron smoke-stack and with advantage to the draft?*

A. R. S.

Answer—The proper method of abating the nuisance from the exhaust steam would be to raise the exhaust pipe 4 or 5 ft. higher than the sides of the building and provide an exhaust-pipe condenser head with a drip pipe returned to a point on the ground where the small amount of vapor from the condensation would be unobjectionable. Discharging the exhaust steam into the smoke-stack would be of little if any advantage in improvement of the draft and, besides causing the interior of the smoke-stack to rust more rapidly, would be likely to result in discharge of moist clots of soot that would be more objectionable than the present annoyance of exhaust steam.

QUESTION [Use of Bent Rocker on Valve Motion]—*Why is a bent rocker used on many high-speed engines for operating the valve rod?* J. E. S.

Answer—With a symmetrical rocker and a valve with equal laps, the port openings will be equal but the piston positions at cutoff will be very unequal and equalization of the cutoff must be at the expense of the lead. By the use of a bent rocker an irregularity may be purposely introduced into the valve motion that may be made to equalize the cutoff without interfering with the equality of the lead, which is of greater importance than equality of port opening.

QUESTION [Increase of Power by Operating Condensing]—*What would be the increase in power of a* 12× 18-*in. engine running* 200 *r.p.m. noncondensing, if operated condensing with* 26 *in. vacuum?* A. B. T.

Answer—A 26-in. vacuum is a pressure $26 \times 0.491 = 12.76$ lb. per sq. in. less than atmospheric pressure. If the back pressure when operating noncondensing is 2 lb. per sq. in. above atmospheric pressure, the reduction of back pressure from operating condensing would be $2 + 12.76 = 14.76$ lb. per sq. in., and the power would be increased

$$\frac{14.76\,(12 \times 12 \times 0.7854) \times 18 \times 2 \times 200}{12 \times 33{,}000} = 30.35 \text{ i.hp.}$$

QUESTION [Economical Limit of Vacuum for Reciprocating Engine]—*Why is more than* 26 *inches vacuum considered uneconomical for operation of a condensing engine?* W. T.

Answer—Under ordinary conditions, when the exhaust is condensed to a pressure below 26 inches vacuum, which would be about 2 lb. absolute and 126 deg. F., the greater cylinder cooling due to the lower temperature of the exhaust, that must be compensated by heat abstracted from the initial steam, amounts to more loss than the increase of m.e.p. obtained by the further reduction of back pressure, aside from additional cost of obtaining a higher vacuum.

QUESTION [Beveled Hub of Cylinder Head]—*Why is the portion of the cylinder head of our engine that projects into the cylinder cut off with a bevel on one side?* A. E.

Answer—The shoulder of the hub for centering the cylinder head is cut away so it will not obstruct connection of the steam passages with the interior of the cylinder. If the cylinder were made long enough to require no such removal of material from the hub, there would be a higher percentage of cylinder clearance volume, thereby resulting in less economical use of steam by the engine.

QUESTION [Forward and Return Strokes]—*Which is the forward and which is the return stroke of a reciprocating engine?* D. A.

Answer—As the piston leaves the head of the cylinder, it is said to be making its forward stroke, and the reverse motion is called the return stroke. The distinction can be fixed in the mind by remembering that with a trunk piston a power stroke is a forward stroke.

QUESTION [Front and Back End and Forward Stroke]—*Which is the front and which is the back end of an engine cylinder and from which end is the forward stroke?* C. R.

Answer—In stationary-engine practice the crank end is called the front end and the opposite end commonly called the head end, is the back end. The forward stroke is toward the front; that is to say, it is the stroke from the back or head end toward the front or crank end. The distinction is readily fixed in mind by referring to the fact that in an engine with a trunk piston, pressure for development of power is exerted on the piston only during its forward stroke, that is, when the piston moves from the back toward the front end of the cylinder. In locomotive practice the end of the cylinder which is in advance when the locomotive moves forward is designated the front end, and therefore the nomenclature is the reverse of that employed in stationary practice.

QUESTION [Advantages of Four-Valve Engine]—*Why is a four-valve engine more economical than a single-valve engine?* W. L.

Answer—Four-valve engines can be designed and constructed with shorter steam passages and less percentage of waste cylinder-clearance spaces than necessary for single-valve engines; and the valves can be adjusted and controlled independently of each other so as to obtain better steam distribution, especially with a variable load, and with separate steam and exhaust valves there is no cooling of the admission valve by the exhaust steam.

QUESTION [Size for Cold-Rolled Steel Head Shaft]—*What size of cold-rolled steel shaft should be used at* 200 *r.p.m. for receiving and transmitting* 100 *hp.?* E. T.

Answer—The formula for cold-rolled head shafts carrying main receiving or driving pulleys and well supported by bearings is $hp. = d^3 \times$ r.p.m. $\div$ 100, in which d = diameter of shaft in inches. For 200 r.p.m. and 100 hp., the formula would become $100 = d^3 \times 200 \div 100$, from which $d = \sqrt[3]{50} = 3.684$, or $3^{11}\!/_{16}$ in.

QUESTION [**Size of Steam Pipe for Engine**]—*What size of steam pipe would be required for an* 18 *in.* $\times$ 36 *in. engine running* 110 *r.p.m., assuming the velocity of steam in the pipe is* 7,000 *ft. per min.?* W. H.

Answer—The piston displacement of the engine will be $(18 \times 18 \times 0.7854) \times 36 \times 2 \times 110 = 2{,}015{,}399$ cu. in. per min. Allowing that the maximum velocity of the steam in the pipe is to be 7,000 ft. or 84,000 inches per minute, when the piston is traveling at its average speed, the required cross-sectional area of the steam pipe would be $2{,}015{,}399 \div 84{,}000 = 23.99$ sq. in. which corresponds to

$$\sqrt{\frac{23.99}{0.7854}} = 5.5. \text{ in. dia.}$$

The least commercial size of pipe required would be 6 in. inside diameter.

QUESTION [**Sufficient Size of Steam Pipe**]—*How can it be known whether the diameter of an engine steam pipe is large enough?* E. N.

Answer—The pipe should be so large that when the engine is carrying its fullest load, the gage pressure at the boiler side of the engine throttle valve will be not less than 97 per cent. of the gage pressure in the outlet side of the boiler stop valve. With steam pipes of ordinary length such a drop of pressure will not be exceeded, allowing for a velocity of steam in the pipe of 6000 ft. per min., a piston speed of 600 ft. per min. and cross-sectional area of steam pipe of about one third the diameter of the engine cylinder.

QUESTION [**Increasing Power Developed by Engine**]—*In how many ways can the horsepower of a non-condensing engine be increased?* C. W. M.

Answer—Assuming that all bearings and working parts are adapted to the changed conditions, the indicated power of an engine is increased for a given m.e.p. by increasing its speed, or by increasing the m.e.p., and the actual or brake horsepower developed for a given indicated power can be increased by reducing the friction of the engine. The m.e.p., can be increased principally by supplying steam of higher initial pressure, by obtaining later and sharper cutoff, by obtaining release later in the stroke, by increasing the percentage of clearance (provided the steam does not follow full stroke), reducing the losses due to cylinder condensation by using a

non-conducting lagging or steam jacketing, or by heating the cylinder without raising the temperature of the exhaust; reducing the back pressure by a freer exhaust or by condensing; and by improving the tightness of the piston, the steam and the exhaust valves. But under some conditions the increase of friction for reduction of leakage may be so great that the actual power will not be increased as much as the indicated.

QUESTION [Forward Pressure, Back Pressure and M.E.P.]—*What is meant by the terms average pressure, mean forward pressure, mean back pressure and mean effective pressure of an engine?* W. P. S.

Answer—The word "mean" is to be understood to signify simple average. The pressure that urges a piston forward in the direction that it travels in performing useful work is called forward pressure, and the average forward pressure is frequently designated as the mean forward pressure. Pressure that opposes the motion of the piston in performance of useful work is called back pressure, and the average back pressure is also called the mean back pressure. The difference between the mean forward pressure and the mean back pressure is called the mean effective pressure.

QUESTION [Horsepower Constant of Engine]—*What is the horsepower constant of an engine having a cylinder* $22\frac{1}{2}$ *in. in diameter and 42-in. stroke with piston rod 3 in. in diameter and running 85 r.p.m.* W. L. G.

Answer—The horsepower constant for each end of the cylinder is the number of indicated horsepower developed by 1 lb. mean effective pressure, and therefore, to express horsepower constant, the usual formula.

$$hp. = \frac{P \times L \times A \times N}{33{,}000}, \text{ becomes } hp.c. = \frac{1 \times L \times A \times N}{33{,}000}$$

By substitution, the horsepower constant for the head end of a $22\frac{1}{2}$ x 42-in. engine, running 85 r.p.m. ,is found to be

$$hp.c. = \frac{1 \times \frac{42}{12} \times 22\frac{1}{2} \times 22\frac{1}{2} \times 0.7854 \times 85}{33{,}000}$$

$$= 3.584 \textit{ hp. per lb. m.e.p. for H end.}$$

Allowing for a 3-in diameter piston rod, the horsepower constant of the crank end is found to be

$$hp.c. = \frac{1 \times \left[\frac{42}{12} \times (22\frac{1}{2} \times 22\frac{1}{2}) - (3 \times 3)\right]}{33{,}000} \times 0.7854 \times 85$$

$$= 3.52 \textit{ hp. per lb. m.e.p. for C end.}$$

The h.p.c. for use with the average of the m.e.p.'s of both ends would be, 3.584 + 3.52 = 7.104 hp. per lb. average m.e.p. of both ends.

QUESTION [Hp. Constant and Required M. E. P.]—*What is the horsepower constant of an 18-in. × 48-in. engine running 74 r.p.m., and what would be mean effective pressure for development of 375 indicated horsepower?* R. W. R.

Answer—The horsepower constant, or power developed per pound of mean effective pressure, is found by the usual formula,

$$Hp. = \frac{PLAN}{33,000}$$

assuming $P = 1$; L = length of stroke in feet; A = area of piston; and N= the number of single strokes. Hence, for the conditions

$$Hp.\ constant = \frac{1 \times \frac{48}{12} \times (18 \times 18 \times 0.7854) \times 74 \times 2}{33000} = 4.565$$

Development of 375 i. hp. would require 375 ÷ 4.565 = 82.15 lb. m. e. p.

QUESTION [Relative Amount of Steam for Instroke and Outstroke]—*Is more steam required by an engine on the instroke or outstroke?* F. S. B.

Answer—No more power is required to be developed by one stroke than the other. With the same percentage of cylinder clearance, same point of cutoff and same compression in each end of the cylinder, there will be less steam required and a development of less power during the instroke on account of the presence of the connecting rod and its reduction of the piston area. This reduction generally amounts to no more than 2 to 3 per cent and, in ordinary operation of horizontal engines, may be neglected. In the case of a vertical engine more steam should be admitted on the upstroke to compensate for lifting the weight of the piston and reciprocating parts during the upstroke and the assistance which their weight renders to the downstroke.

QUESTION [Water Rate of Engine]—*What is meant by an engine running on 12 lb. of water per horsepower-hour?* J. E.

Answer—The statement is intended to signify that the weight of steam consumed is 12 lb. per horsepower-hour, which would be the weight of steam generated from 12 lb. of water or the "water-rate." A statement of water rate is incomplete in expressing economy of a steam engine unless also the pressure and quality of steam supplied and the back pressure on the exhaust are specified or understood.

QUESTION [Percentage of Fuel Energy Realized by Engine]—*An engine uses 25 lb. of steam per indicated horsepower-hour, and the evaporative economy of the boiler, under the*

operating conditions, is 8 lb. of water per pound of coal. Assuming that the heat value of the coal is 10,000 B.t.u. per pound, what percentage of energy contained in the coal is realized by the engine? R. L.

Answer—The fuel consumption would be 25 ÷ 8 = 3.125 lb. of coal per horsepower-hour, and as 1 hp.-hr. = 33,000 × 60 = 1,980,000 ft.-lb., there would be 1,980,000 ÷ 3.125 = 633,600 ft.-lb. realized per pound of the coal. One B.t.u. is equal to 777.5 ft.-lb., and if the coal contains 10,000 B.t.u. per pound, the energy in a pound of the coal would be 7,775,000 ft.-lb., and the energy realized by the engine per pound of coal used would amount to 633,600 × 100 ÷ 7,775,000 = 8.15 per cent.

QUESTION [**Ratio of Expansion**]—*What is meant by the ratio of expansion in an engine?* W. L. S.

Answer—In the operation of a steam engine, the ratio of expansion is the number of times the steam is expanded after cutoff occurs. If there is three cubic feet of steam present at cutoff and this is expanded to nine cubic feet, or three times the volume, the ratio of expansion is 3. The approximate ratio of expansion is the reciprocal of the fraction of stroke at which cutoff occurs. Thus, if cutoff occurs at $\frac{1}{4}$ stroke, the approximate ratio of expansion is 3. But the actual volume at cutoff and after expansion includes the clearance volume at one end of the cylinder and must be included to find the actual ratio of expansion. Thus, if the clearance is 5 per cent of the piston displacement, and cutoff occurs at 0.33 of the stroke, the actual ratio of expansion would be

$$\frac{1 + 0.05}{0.33 + 0.05} = 2.76$$

QUESTION [**Cylinder Ratios of Compound Engines**]—*Is it considered practical to construct and operate compound engines with a cylinder ratio as high as 6 to 1?* J. P. O.

Answer—In a cross-compound engine it is desirable to have the load divided equally between the cylinders and then make the cylinder ratio such as to sustain the least drop of pressure between the cylinders and the least loss from range of temperatures in the cylinders. For present-day boiler pressures of 100 to 200 lb., this can be accomplished with cylinder ratios between $4\frac{1}{2}$ and 5 with variation of load between one-half and full load. Ratios between 6 and 7 have been used on special engines with good economy when operated with loads and steam pressures appropriate for the ratio. These engines were marine and water-works engines carrying uniform loads. For stationary cross-compound engines with variable loads, the smaller cylinder ratios have given better results.

QUESTION [**Allowance for Moisture in Rating Engine Economy**]—*In determining the economy of reciprocating*

steam engines expressed in pounds of steam per indicated horsepower-hour or per brake horsepower-hour, how is allowance to be made for the quality of the steam supplied? A. E. M.

Answer—Unless otherwise particularly specified, reciprocating-engine economy is assumed to be based on a supply of commercially dry steam, that is, steam containing not over 2 per cent moisture, and the quantity of steam charged as consumed is determined by deducting the moisture, if any, found by calorimeter test, from the total amount of air-pump discharge or complete condensation of exhaust discharged. If the steam is superheated, no correction is to be made for the superheat. When the steam supply is of poorer quality than 98 per cent dry, the moisture is more detrimental to economy than the percentage present, on account of reduction of the expansive force of the dry portion of steam by heat absorbed in evaporation of the moisture, and also because the losses from radiation are increased owing to greater conductivity of heat by wet steam than from dry steam coming in contact with the walls of the cylinder and steam passages.

QUESTION [Effect of Moisture on Engine Economy] —*What effect has moisture of steam on economy of an engine? An engine that is guaranteed to develop a horsepower-hour on* 28 *lb. of dry steam is found to require* 30.8 *lb. of steam containing* $5\frac{1}{2}$ *per cent. of moisture. What per cent. is the performance above or below the guarantee?* R. N. L.

Answer—Experiments on engine economies, using steam of various percentages of dryness, seem to confirm the fact that the ill effect of moisture is proportional to the percentage of moisture present; that is, the consumption of dry steam remains practically constant. Accordingly, a consumption of 30.8 lb. of steam containing $5\frac{1}{2}$ per cent. moisture would be equivalent to consumption of $30.8 \times (1 - 0.55) = 29.1$ lb. of dry steam, and if 28 lb. of dry steam was guaranteed, it is fair to assume that the steam consumption is $(29.1 - 28) \div 28 = 0.039$ or practically 4 per cent. greater than guaranteed.

Section III
Boilers

QUESTION [Measuring Length Required for New Boiler Tubes]—*What allowance should be made in cutting the length of new tubes for a return-tubular boiler?* L. F.

Answer—There is more or less irregularity in the tube heads of a boiler, and measurements should be made for each tube separately by rodding the distance from outside to outside of the tube heads with a stiff wooden rod or gas pipe, small enough to pass through the tube holes. The measuring rod or pipe should be held flush with one tube head by a helper and marked close against the outside of the other head. When this outside distance between tube heads has been set off on the tube from one end, enough extra length should be measured off to allow for each end from two to two and one-half times the thickness of the tube, for beading.

QUESTION [Coefficient of Expansion of Firebrick]—*What is the coefficient of expansion of firebrick?* F. E. C.

Answer—Experiments have shown that the coefficient of expansion of firebrick decreases with temperature rise, but an average coefficient of linear expansion within the range of temperatures ordinarily attained in boiler furnaces would be about 0.0,000,027 per degree F.

QUESTION [General Tests of Hardness and Acidity of Feed Water]—*What simple tests can be made to determine the hardness and the acidity of boiler-feed water?* H. S. D.

Answer—Hardness is shown by adding to a sample of the water a few drops of a solution of castile soap in alcohol. If hard the water will be made a milky white, and if soft the water will remain clear. An acid in the water will turn blue litmur paper red. Water which will turn blue litmus paper res before the water is boiled but not after boiling contains cadbonic acid.

QUESTION [Safety Valves for Superheaters]—*Should superheaters have safety valves attached?* E. S.

Answer—All superheaters should be equipped with safety valves, which should be set slightly lower than the boiler valves. With the superheater valves blowing first, a flow of

steam through the superheater is assured should the load be suddenly thrown off the boiler, and there will be no danger of burning superheater tubes, as would be the case were the boilers valves to blow first. Again, the use of safety valves on the superheater outlets will prevent highly heated steam passing to the auxiliaries should the main power units be suddenly shut down.

QUESTION [Loss of Draft Through Boiler]—*Wha would be a fair allowance for loss of draft through a water-tube boiler?* R. G.

Answer—The draft resistance varies largely with the design of the boiler and the percentage of capacity at which the boiler is operated. In a well-designed water-tube boiler the loss is about 0.25 in. of water when the boiler is operated at rating, 0.4 in. at 150 per cent of rating, and 0.65 in. at 200 per cent of rating

QUESTION [Variation from Test Gage]—*If a boiler steam gage registers 2 lb. higher than a test gage at 20 lb., will it register 2 or more pounds higher than the test gage at 100 pounds?* E. M.

Answer—Because a gage is a certain amount ahead or behind a test gage at one point on the scale, it does not follow that it will be ahead or behind the test gage at other points of the scale, nor the amount of the variation.

QUESTION [Steam Required for Heating Water]—*What quantity of steam at 100 lb. gage would be condensed when discharged into 400 gal. of water contained in an open tank, to raise the temperature of the water from 40 deg. to boiling?* H. M.

Answer—The initial 400 gal. of water would weigh $400 \times 8\frac{1}{3} = 3{,}333$ lb. and neglecting losses of heat from radiation, to raise the temperature from 40 deg. F. to the temperature of boiling at atmospheric pressure would require $3{,}333 \times (212 - 40) = 573.276$ B.t.u. Assuming that dry saturated steam is upplied at the pressure of 100 lb. gage or 115 lb. per sq. in. abs., then each pound (weight) of the steam would contain 1,188.8 B.t.u. above 32 deg. F. and, in becoming condensed to 212 deg. F. would part with $1{,}188.8 + 32 - 212 = 1{,}008.8$ B.t.u.; and since heating the water would absorb 573,276 B.t.u., the amount of steam condensed would be $573{,}276 \div 1{,}008.8 = 568.3$ lb.

QUESTION [Pipe Threads for Boiler Connections]—*What is the rule for the number of screw threads there should be for a tapped pipe connection to a boiler?* R. L.

Answer—Threaded openings in a boiler for pipe connections 1 in. in diameter or over should have not less than the number of perfect threads required by the A. S. M. E. Boiler Code, as per the following table:

MINIMUM NUMBER OF PIPE THREADS FOR CONNECTIONS TO BOILERS

Size of pipe connection, in.	1 and 1¼	1½ and 2	2½ to 4 inclusive	4½ to 6 inclusive	7 and 8	9 and 10	12
Number of threads per in.	11½	11½	8	8	8	8	8
Minimum number of threads required in opening	4	5	7		10	12	13
Minimum thickness of material required to give above number of threads, in.	0.348	0.435	0.875	1	1 25	1.5	1.625

When the boiler material is not of sufficient thickness the obtain the designated number of standard pipe threads, thero should be a pressed steel flange, bronze composition flange, steel-cast flange, or steel plate, constructed and riveted to the boiler so as to give the required number of perfect threads.

QUESTION [Interpretation of Safety-Valve Formula] —*Where the formula for the size of safety valve for a boiler is* $A = W \times 70 \times 11 \div P$, *in which A equals the area of direct spring-loaded safety valve in square inches per square foot of grate surface, W equals the weight of water in pounds evaporated per square foot of grate surface per second, and P equals the pressure (absolute) at which the safety valve is set to blow, how would the formula be adapted to the use of oil as fuel?* W. E. T.

Answer—As in the original formula A represents the area of safety valve per square foot of grate surface, then

$$\text{Total area of safety valve} = \text{number of sq.ft. of grate} \times W \times 70 \times 11 \div P$$

But as square feet of grate $\times$ W = total weight of water evaporated per second, and as evaporation per pound of oil $\times$ number of pounds of oil burned per second would amount to the same thing, then for oil burning, the formula for area of safety valve would become,

Area of spring-loaded safety valve in square inches = evaporation per pound of oil $\times$ *number of pounds of oil burned per second* $\times 70 \times 11 \div P$.

QUESTION [Required Height of Stack]—*What is the formula and computation to determine the required height for a stack* 80 *in. in diameter to serve* 1,000 *hp.?* W. O. A.

Answer—No stack formula can be constructed to fit all cases exactly, as too many varying factors enter each case. However, an empirical formula for height and diameter of stack that is largely used by engineers is that proposed by William Kent; namely,

$$\text{Hp.} = 3.33\,(A - 0.6\sqrt{A})\,\sqrt{H}$$

where Hp. is the number of horsepower served, based on a coal consumption of 5 lb. per rated horsepower per hour; A is the actual cross-sectional area of the flue in square feet, and H is the height of chimney above the grate in feet.

The quantity $(A - 0.6\sqrt{A})$ = the effective area of stack, based on the assumption that the retarding effect by friction of the ascending gases may be considered as equivalent to the diminution of the area of cross-section of the flue equal to a layer of gas 2 in. thick over the whole interior surface. Taking Hp. = 1,000 and $A = \left(\frac{80}{12}\right)^2 \times 0.7854 = 34.9$ square feet then $A - 0.6\sqrt{A} = 34.9 - 0.6\sqrt{34.9} = 31.36$ square feet. Substituting the formula becomes $1{,}000 \doteq 3.33 \times 31.36 \times \sqrt{H}$, from which the value of H is found to be about 92 feet.

QUESTION [Heat Radiated from Boilers]—*In boiler practice what percentage of heat is lost by radiation to the atmosphere?* E. B.

Answer—The losses by direct radiation vary with the kind and size of boilers, the percentage of rated capacities at which they are operated and the surrounding air temperatures to which the boilers are exposed. Radiation loss usually amounts to 12 to 15 per cent for small boilers operated at low rates, 7 to 8 per cent for fair-sized units at average loads and 2 to 3 per cent for very large boilers operated at high percentages of rating.

QUESTION [Variation of Water in Gage Glass]—*Why does the water rise and fall in the glass gage of a boiler when there is no variation in draft of steam or rate of feeding water to the boiler?* E. E. R.

Answer—The water rises and falls in the glass gage because there is a variation of pressure in the boiler at the point where the water column is connected. This may be due to circulation of the water, or to variable rarefaction of the water from irregular formation of steam bubbles, or a surging of the water from being swept along by the steam bubbles in their ascent to the steam space of the boiler.

QUESTION [Operation of Holly Steam Loop]—*In the Holly steam loop how is the condensation in the bottom of the receiver vaporized so it may be forced up into the receptacle placed on the roof of the building, whence it falls into the drop leg?* T. J. R.

Answer—In operation of the Holly loop the condensation from pipes, traps and separators gravitates to the receiver,

from which it is forced into the "riser" in the form of spray. The spraying effect is produced by forcing the condensate from the receiver through a series of small holes drilled in the sides of a short vertical pipe inside of the receiver that is connected to the bottom of the receiver like a standing overflow and that discharges through an outlet in the bottom of the receiver. From this receiver the spray and moisture rise to the "discharge chamber," or receptacle placed on the roof of the building, on account of the lower pressure at that point, where the steam and entrained water are separated. The water thence falls by gravity to the drop leg, and the steam is discharged through a reducing valve into the feed-water heater.

QUESTION [**Cost of Power in Isolated Plants**]—*What is a fair figure for the cost of power in a factory plant?*

M. S. T.

Answer—A recent number of the *Sibley Journal* contained an article on Power Costs, 1914-19, by Hubert E. Collins, in which appeared comparative data applying to a number of power plants between the years indicated. One of these, situated in the Mohawk Valley, New York State, gives records quite complete in detail, which are reproduced herewith. The fuel used was anthracite dust and bituminous mixed in equal proportions for the years 1914 to 1916 inclusive and bituminous only in 1917 and 1918. Induced draft was used in 1914 and 1915 and natural draft in 1916-18. Fuel costs were based on long tons delivered to siding.

TABLE I. COMPARISON OF POWER PLANT PERFORMANCES

	1914	1915	1916	1917	1918
Tons coal used	9,208	9,171	11,164	12,469	12,739
Cost of coal	$26,045	$21,333	$31,894	$37,533	$40,786
Pounds coal	20,655,920	20,544,160	25,007,360	27,932,128	28,536,928
Pounds steam production	181,772,096	169,136,636	195,256,860	204,189,600	206,106.150
Kw. hr. production	1,561,318	1,623,153	2,036,5C7	1,981,370	1,970,970
B.hp. production	6 059,066	5,717.520	6.508,562	6,806 320	6,870,205
Working days	277	292	292	292	297
Heating hours	3,080	3 024	3,182	3,361	3,423
Average evaporation	8.8	8.5	7.8	7.3	7 2
Water per kw.-hr	21.9	28.0	22.0	22.8	22.8
Coal per kw.-hr	2.49	3.3	2.8	3.1	3.1
Coal per b.-hp	3.4	3.5	3.8	4.1	4 2
Cost of coal per pound	0 00126	0.00105	0.00128	0.00134	0.00143
Cost of coal per b.-hp	0.0042	0.0037	0.0048	0.0055	0.006
Cost of coal per hw.-hr	0.00313	0.0035	0.00358	0.0045	0.00443

TABLE II. AVERAGE COST PER KILOWATT HOUR

	1914	1915	1916	1917	1918
Wages	$0.00159	$0.00147	$0.00147	$0.00168	$0 00198
Water	.000131	.000168	.00013	.000137	.000189
Fuel	.00313	.0035	.00358	.0045	.00443
Dep. on building	.00127	.00123	.00098	.001	.001
Dep. on machinery	.00338	.00325	.0026	.00267	.00267
Total cost	$0.009501	$0.009618	$0.00876	$0.009987	$0.010269

QUESTION [**Relative Effect of Low- and High-Pressure Radiators**]—*What is the relative effectiveness of dry room*

radiators supplied with exhaust steam at a pressure of 2 lb. gage and when supplied with live steam at 100 lb. gage? B. C.

Answer—When there is complete removal of air from the radiators and perfect circulation of steam, the heat radiated for a square foot of surface may be considered as directly in proportion to the difference of temperature of the steam within the radiator and the room temperature in the immediate vicinity of the radiator. Assuming, for instance, that the room temperature maintained near the radiator is 90 deg. F., as the temperature of steam at 2 lb. gage is 218.5 deg. F., and the temperature of steam at 100 lb. gage is 338 deg. F., then the relative effectiveness per square foot of radiation would be as (218.5 − 90) for the exhaust steam, to (338 − 90) for the steam at 100 lb. gage, or as 100 to 193.

QUESTION [How to Figure the Weight of Any Gas]-*Is there a formula by which one can find the weight per cu.ft. of a gas?* W. N. P.

Answer—It is convenient to know the constant by which the weight of a cubic foot of any gas may be approximately determined from its molecular weight. At atmospheric pressure (14.7 lb. absolute) and 60 deg. F. the relation is as follows:

Weight per cubic foot = *molecular weight* ÷ 376

Cubic feet per pound = 376 ÷ *molecular weight*

The molecular weights of a few common gases are as follows: Nitrogen (N_2), 28; Oxygen (O_2), 32; Carbon Dioxide (CO_2), 44; Carbon Monoxide (CO), 28; Hydrogen (H_2), 2; Ammonia (NH_3), 17. For air use 28.8 (a weighted average based on four parts nitrogen and 1 part oxygen).

As an example, the weight of a cubic foot of CO_2 at 14.7 lb. absolute pressure and 60 deg. F. would be 44 ÷ 376 = 0.117 lb., and the number of cubic feet in a pound would be 376 ÷ 44 = 8.55.

QUESTION [Heating Surface of Iron Pipe]—*What amount of heating surface should be allowed per running foot of iron pipe of sizes generally used for steam-heating coils?* C. B. F.

Answer—Coils for steam heating usually are made from pipe having nominal diameters of ¾ in. to 2 in. The heating surface of these sizes is as follows:

1 lin.ft. of ¾-in. pipe has 0.2748 sq.ft. heating surface
1 " " " 1 -in. " " 0.3442 " " " "
1 " " " 1¼-in. " " 0.4345 " " " "
1 " " " 1½-in. " " 0.4975 " " " "
1 " " " 2 -in. " " 0.6218 " " " "

QUESTION [Preheated Air]—*Is an air preheater justified in a medium-sized boiler house?*

Answer—The air recovers part of the flue-gas heat and so increases the efficency. If the plant is fairly well loaded, so that the flue-gas temperature is high, the preheater might be economically profitable.

QUESTION [Independent Stacks for Horizontal Return-Tubular Boilers]—*For a suburban factory power plant, begun with the installation of two horizontal return-tubular boilers that probably will be duplicated within a year, would it be better to supply each boiler with a separate steel stack set over the uptake, or provide a steel stack that will be adequate for all boilers?*

J. B. N.

Answer—The best draft control would be obtained by furnishing each boiler with a separate stack, but placing the stacks directly over the boiler uptakes is objectionable on account of the expense of providing supports suitably independent of the boiler settings and trouble will be experienced from soot and scale dropping from the inside of the stacks into the uptakes of the boiler. The arrangement also would be likely to give trouble from rain water running down the sides of the stacks and finding its way to the boiler settings. These disadvantages can be obviated and nearly the same draft advantages of independent stacks can be secured by setting the stacks at the sides of the settings or to the rear of the firing spaces, and providing the connections from the uptakes to the separate stacks with easy bends. For most situations the independent stack supports and connections should cost no more than proper provision for independently supporting the stacks directly over the uptakes of the boilers.

QUESTION [Size of Flue for Return-Tubular Boilers]—*What should be the size of a circular steel flue chimney connection 60 ft. long for two 72 in. diameter x 18 ft. return-tubular boilers?*

W. N. F.

Answer—A safe rule to follow in figuring flue areas is to allow 5 sq. in. of cross-sectional area per boiler horsepower, and assuming that 300 boiler horsepower is to be provided for, the cross-sectional area should be 1,500 sq. in., or practically 44 in. diameter.

QUESTION [Provision for Smoke Connection to Stack]—*A return-tubular boiler that has thirty-six 3-in. fire tubes has the grate area reduced to 6 sq. ft. What size of smoke connection should it have to a 40-ft. chimney that has a 2 x 2 ft. flue?*

C. E. S.

Answer—A 12-in. diameter smoke pipe with easy bends at angles would be sufficient for the present grate area. But

to provide for future enlargement of the grate, with maximum draft area of the tubes, it would be well to supply a stack connection about 18 in. in diameter, in any case providing a damper in or near the smoke uptake of the boiler.

QUESTION [Height of Closing-in Side Walls of H. R. T. Boiler Setting]—*At what height should the side walls of a horizontal return-tubular boiler setting be closed in against the sides of the boiler?* A. T.

Answer—The furnace sides of the walls should be closed in no higher than the highest row of tubes, and for durability of the setting it is preferable to close in not higher than the center of the boiler. Any heating surface obtained of the shell above the center of the boiler receives little benefit of circulation of the heated gases and soon becomes valueless on account of deposits of soot and ashes.

QUESTION [Hollow Walls for Boiler Settings]—*What is the advantage or disadvantage of hollow walls for boiler furnaces?* F. A.

Answer—So far as direct loss of heat is concerned, there is rather more heat lost by radiation through air-space walls than through solid walls of the same total breadth, especially if the air space is near the furnace side, as air offers less resistance to radiation than brickwork. A hollow wall is better than a solid wall to prevent cracks from expansion of the brickwork on the furnace side, but when this construction is adopted, the cavity should be filled with solid loose material to prevent radiation of heat from the inner to the outer wall and retard circulation of cold air entering the cavity through openings or cracks that may occur in the outer wall.

QUESTION [Sizes of Tubes for Return-Tubular Boilers]—*What determines the sizes of tubes suitable for a horizontal return-tubular boiler?* W. A.

Answer—The most suitable size of tubes depends on their length, the kind of coal to be burned, the facilities for cleaning and the force of the draft. The smaller the tubes the greater the total heating surface for the space occupied, but the more easily they clog with soot and choke the draft. The larger the flues the less the heating surface for a given amount of tube space and less efficiency of heating surface. For operation with natural draft a common rule is to allow one inch of nominal diameter for each four feet of length for use of bituminous coal or for each five feet of length for anthracite. With forced draft and frequent cleaning smaller tubes can be used.

QUESTION [Beading Tube Ends of Fire Tube Boilers]—*How is the holding power of tubes of fire tube boilers affected by beading-over the ends?* R. A.

Answer—The operation of beading over the tube ends has a tendency to drive the expanded tube from the hole in the tube sheet and thus impair the tightness of the joint and excessive beading produces granulation and reduction of strength of the material. But when the operation of beading is carefully performed on good tube material, the holding powers of beaded and flared tubes are about the same, with the advantage of greater durability of beaded tube ends over those that are simply flared in resisting the burning action of the fire.

QUESTION [**Handholes for Vertical Fire-Tube Boilers**]—*How many handholes should there be in a vertical fire-tube boiler?* G. L. S.

Answer—Boilers more than 24 in. in diameter should have not less than seven handholes; namely, three in the shell at or about the line of the furnace crown sheet, one in the shell at or about the water line or opposite the fusible plug when used, and three in the shell at the lower part of the water leg.

QUESTION [**Advantages of Diagonal Over Head-to-Head Stays**]—*What are the advantages of diagonal stays over head-to-head stays in a return-tubular boiler?* W. N. E.

Answer—The shell has sufficient strength to resist all the pressure on the areas of the heads required to be stayed above and below the tubes, and that strength can be utilized by the employment of diagonal stays with the advantage of occupying comparatively little space in the steam room, thus permitting easier inspection and cleaning than when through bolts are used; and diagonal stays being shorter than through stays, there is less effect from unequal expansion.

QUESTION [**Unstayed Heads of Steam Drums**]—*Why are not heads of steam drums of water-tube boilers stayed?* J. N. R.

Answer—When unstayed, the heads are bumped. When a head is bumped or dished to the form of a spherical segment, most of the material is subjected only to a tensile stress such as would occur if the segment were part of a complete hollow shpere of the same radius of curvature and subjected to the same pressure, without requirement of stays to preserve a spherical form. However, when the pressure is on the concave side and the head is not a complete hemisphere, the pressure tends to distort the material to a hemispherical form at the periphery where the original form is a sudden change of shape from the spherical form to the cylindrical form of the flange. This gives rise to a dangerous bending action which must be prevented by staying or more usually is met by the thickness of the material. To guard against this weakness, where the heads are assumed to be of the same thickness throughout, conservative formulas for determining the thickness of unstayed bumped heads for a given working pressure result in

providing greater thickness of material than would be necessary for obtaining sufficient strength for the central spherical portion of the head.

QUESTION [Sling Stays of Internally Fired Boilers]—*What are sling stays, used in construction of internally fired boilers?* R. G.

Answer—In locomotive and marine types of boilers, where girder stays are used for supporting the crown sheet or top of the combustion chamber, the girder stays, in addition to having end supports, are commonly supported at intervals of their span by short articulated stays, called sling stays, fastened to the girder and top of the boiler. The required depth of the girder stays is thereby reduced, and with the crown sheet suspended from the girders the staying is equivalent to having a flexible arrangement of through stays from the crown sheet to the top sheet of the boilers.

QUESTION [Reduction of Plate Thickness at Circumferential Joints]—*Why is it necessary to plane down that part of boiler plates forming the laps of the circumferential joints of horizontal return-tubular boilers when the thickness of plates exceeds 9/16 in.? Does this not reduce the strength of the joint?* P. F.

Answer—The portion of the plates forming the laps of the circumferential joints where exposed to the fire or products of combustion, should be planed or milled down to ½ in. in thickness to render the plate thin enough for the plates and rivets at the joints to be kept sufficiently cooled by the water of the boiler to prevent their becoming burned. While this reduction of thickness lowers the strength of the joint, it may be reduced to only 35 per cent. of the strength of the longitudinal joints, provided 50 per cent. or more of the load that would act on an unstayed solid head of the same diameter is relieved by the effect of tubes or through stays, and in that way plates of unusual thickness and longitudinal joints of highest efficiency may be employed with circumferential joints planed down to ½ in. in thickness without reduction of working pressures below those appropriate to the strength of the longitudinal joints.

QUESTION [Detecting Fractured Staybolt]—*What is the best method of determining whether staybolts in the side sheets of a locomotive-type boiler are defective?* L. V.

Answer—When such staybolts become fractured the rupture generally occurs near the outside shell, and by drilling a small hole in the staybolt extending from its outside end to a point that is about ¾ in. beyond the inside surface of the outer sheet a fracture will be exposed by the discharge of water and steam.

QUESTION [Size of Uptake for Horizontal Return-Tubular Boiler]—*What should be the size of smoke uptake for a 72-in. x 18-ft. horizontal return-tubular boiler?* W. L. M.

Answer—The uptake area is usually made one-eighth the area of the grate, although, with tight connections and a tall stack this figure may be considerably reduced. However, it is well to provide for extreme conditions and allowing for 40 sq. ft. of grate area, and one-eighth grate area for that of the uptake would give 5 sq. ft., or 720 sq. in. for the cross-sectional area of the uptake.

QUESTION [Relative Pressure for Convex and Concave Heads]—*What is the relative strength of convex and concave head for a steam drum?* B. J. T.

Answer—A concave head, or one dished inward so the drum pressure shall come on the convex side, should be used for a maximum working pressure of only 60 per cent. of that allowed for a head of the same dimensions with the pressure on the concave side.

QUESTION [Advantage of Inclining Hand-Fired Grate]—*For a hand-fired boiler what is the advantage of having the rear end of the grate lower than the front end?* J. A. A.

Answer—The draft is stronger in the rear end of the ashpit, and with even firing a thicker fire becomes built up at the rear end of the grate; and having the grate pitched toward the rear end makes slicing and firing easier.

QUESTION [Grooving of Boiler Shells]—*What is grooving of a boiler shell, and by what is it caused?* F. R.

Answer—Grooving is internal corrosion or rusting away of the boiler shell in the form of grooves or cavities. Corrosion of this kind generally may be attributed to local changes of shape of the boiler shell due to changes of pressure or temperature. Slight movements loosen particles of skin or rust, and fresh surfaces are exposed for the formation of more rust. Corrosion, thus forming grooves or cavities, proceeds rapidly in such places as at sharp corners of head flanges, in the lower part of the water legs of locomotive and of vertical boilers, especially when fitted with mud rings, and along the side seams of cylindrical shells that are made with lap joints as a result of the bending that occurs in a lap seam when under pressure.

QUESTION [Sloping Floor of Combustion Chamber]—*In the setting of a horizontal return-tubular boiler, what is the advantage of having the floor of the combustion chamber extended with a slope downward from the upper part of the bridge wall to the rear of the setting?* D. M.

Answer—There is no advantage in sloping the floor beyond obtaining greater convenience for drawing soot and cinders with a hoe toward the rear of the setting when cleaning out

the combustion chamber. It is desirable to have thorough mixture of the gases before they reach the return connection of the setting, and for that purpose more favorable eddying of the gases is obtained by having the combustion chamber begin with a deep drop below the top of the bridge wall.

QUESTION [Connecting Water Column of R. T. Boiler]—*How should a water column be connected to a return-tubular boiler?* R. M. P.

Answer—The pipes connecting the water column to the boiler should be not less than 1 in., with the steam connection taken from the top of the shell, or the upper part of the head, and the water connection should be of brass pipe taken from a point not less than 6 in. below the center of the shell and provided with plugged cross fittings at angles to facilitate cleaning. The water column, or water connection, should be provided with a drain cock or drain valve with suitable connection not less than $\frac{3}{4}$ in. to the ash pit or other safe point of waste. Both the water and steam connections should be provided with a shutoff consisting of either outside screw and yoke type gate valve or stop cock with permanent lever and such valves or cocks should be locked or sealed open and located as close as practicable to the boiler. The water column should be provided with water glass stop and drainage fittings set at such height that the lowest visible part of the glass gage will be not less than 4 in. above the upper row of tubes, and unless there are two water columns located on the same horizontal line and not less than 2 ft. apart, the water column should be provided with at least three gage cocks located within the range of the visible length of the water glass, unless the boiler is otherwise provided with the same number of gage cocks located at similar levels.

QUESTION [Equal Width Straps for Boiler Joints]—*What are the advantages and disadvantages of having the same width for the inside and outside straps of a longitudinal butt and double-strap boiler joint?* E. L. L.

Answer—When both straps are of the same width, each held with the same number of rows of rivets, there is an advantage over unequal width straps in obviating eccentricity of shearing stress of the rivets due to unequal width of straps which has a tendency to bend the shell out of true circular form and thereby introduce longitudinal fracture of the main plate along the edges of the straps.

In American practice this disadvantage of unequal-width straps is assumed to be overcome by the stiffening effect of the broader inside strap. The principal disadvantages of employing straps of equal width are that the joints cannot be designed of as high percentage of efficiency, since for the best efficiency the outer rows of rivets must be spaced too far apart to form a good calking pitch. The latter objection can be overcome only by the expensive process of making the lacking

edges of saw-tooth form, with rounds and fillets that are more difficult to calk, and the calking edges must be of considerably greater length than straight calking edges.

QUESTION [Direction of Laps of Horizontal R. T. Boilers]—*Why do the inside laps on longitudinal seams of horizontal return tubular boilers point downward?* H. S.

Answer—The laps are made so the outside lap will point upwards and may be calked with the greater convenience of "down hand" calking.

QUESTION [Allowance for Strength of Girth Seams] —*In computing the strength required of girth joints of a horizontal return tubular boiler shell, what allowance should be made for the reduction of area of the heads due to the tubes?* W. L. C.

Answer—According to the A. S. M. E. Boiler Code, par. 184, b: "When 50 per cent or more of the load which would act on an unstayed solid head of the same diameter as the shell, is relieved by the effect of tubes or through stays, in consequence of the reduction of the area acted on by the pressure and the holding power of the tubes and stays, the strength of the circumferential joints in the shell shall be at least 35 per cent of that required for the longitudinal joints." When less than 50 per cent of the solid head area is thus relieved, the strength of circumferential joints should be at least one-half the strength required for the longitudinal joints.

QUESTION [Relative Stresses of Girth and of Longitudinal Seam]—*Why is the girth seam of a boiler twice as strong as the longitudinal seam?* N. R.

Answer—The strength of a joint is no greater because it is used as a girth seam, but the stress pulling the joint apart is only one-half as much per unit length of a girth seam as per unit length of a longitudinal seam. As an example, suppose a cylindrical shell to be 72 in. in diameter. Then the whole length of a girth seam would be $72 \times 3.1416 = 226.195$ in. The gross area of a head 72 in. in diameter would be $72 \times 72 \times 0.7854 = 4071.5136$ and each pound pressure per square inch would cause a stress of $(1 \times 4071.5136) \div 226.195 = 18$ lb. per inch length of the girth seam; but for 1 in. length of a longitudinal seam, each pound pressure per square inch would cause a stress of $\frac{1}{2}$ of $(1 \times 72) = 36$ lb., or twice as much as the stress per inch of length of the girth seam.

QUESTION [Girth Seams Lapped Toward Fire]—*What is the disadvantage of having the ring seams of a horizontal return-tubular boiler lap toward the fire?*

Answer—The outer plate of the lap is more or less overheated and has a tendency to expand away from the inner plate and cause leakage, and more especially when the lap is toward the fire, as the calking is exposed to the direct impingement of the heated gases.

QUESTION [Triple Riveting of Lap Joints]—*Why are lap joints of high-pressure steam drums frequently made with three or more rows of rivets?* F. C.

Answer—A triple-riveted lap joint may be designed of higher efficiency than a double-riveted lap joint, and the greater width over the rows of rivets renders it easier to bring the plates into line, with the result that there is less bending than in the single-or double-riveted lap joint.

QUESTION [Single Shear and Double Shear]—*What is meant by a boiler rivet being in single shear or in double shear?* F. A. P.

Answer—A rivet is said to be in single shear when it is subject to shearing action that tends to produce cleavage at a single cross-section of the rivet, as when used in a riveted lap-joint for holding together two plates that pull in opposite directions. A rivet is said to be in double shear when subject to shearing action that tends to produce cleavage at two cross-sections of its length, as when the rivet is used in a riveted butt-and-double-strap joint for holding together a main plate, sandwiched between an inside and outside cover plate whose direction of pull is opposite to that of the main plate.

QUESTION [Testing Tightness of Boiler Rivet]—*How can it be determined whether a boiler rivet is driven tight?* S. M. E.

Answer—To test the tightness of a rivet, place the thumb against one side of the rivet head and the forefinger of the same hand on the plate. Then, by striking the rivet head with a hammer, any looseness of the rivet may be felt.

QUESTION [Pitch of Rivets of Girth Seams]—*Why is the pitch of rivets on girth seams of horizontal return-tubular boilers generally made less than in the longitudinal seams?* W. P.

Answer—The outer plate in a girth seam is more or less over-heated where the seam passes over the fire, and there is a tendency for the outer plate to expand away from the inner one, resulting in leakage. For this reason the space between rivets is made smaller than on cool seams of the boiler.

QUESTION [Single Riveting of Vertical Boiler Furnace Sheets]—*Why is single riveting considered to be sufficient for the furnace-sheet joints of large vertical boilers?* J. J. H.

Answer—The shell of the firebox is subjected to a collapsing pressure which is resisted by staybolts that pass through the water leg to the outer shell, and when the staybolts have proper strength and spacing, the riveting of the longitudinal joint of the firebox sheet has little more to perform than to hold together the laps of the joint for calking.

QUESTION [Slippage of Riveted Joint of Boiler]—*Is slippage of a riveted boiler joint an indication that the joint has been stressed to the point of failure?* F. M.

Answer—In a newly constructed joint the resistance to initial slippage consists mainly of friction between the plates due to the longitudinal shrinkage of the rivets from cooling after they are driven. Some displacement of the plates must occur before the sides of the holes are brought to a bearing to exert a shearing action on the rivets, because the rivets in cooling also contract in diameter and do not completely fill the rivet holes. In American boiler practice the assistance that is rendered by friction between the plates is regarded as uncertain and the strength of riveted joints is based on resistance to stresses concomitant with shearing action on the rivets; and sufficient slippage for taking up bearings on the rivets that have been properly driven is only an indication that the rivets are in position for receiving the shearing stresses, and not that the joint has been stressed to the point of failure.

QUESTION [Splitting of Sheet at Rivet Holes of Girth Seam]—*What causes cracks from the rivets to the edge of the sheet in the girth seam of a horizontal return tubular boiler, immediately over the fire?* H. M.

Answer—The texture of the material may have been injured in the construction of the boiler from driving drift pins or from setting the rivets too cold, but more often, splitting at the edge of the sheet is due to expansion of the rivets from becoming overheated, especially when there is scale in the boiler.

QUESTION [Repair of Leaky Girth Seams Having Fire Cracks]—*We have two horizontal return tubular boilers that leak at the girth seams, caused by fire cracks. I have tried calking and replaced the old rivets with larger ones driven in reamed holes, but cannot stop the leaks. I want to prolong the life of the boilers. What is the remedy for the trouble?* G. B.

Answer—If the fire cracks at the points of leakage extend clear through the outside laps, or the material has been so badly burned as to render the texture of the material unsuitable for calking after cutting away the burnt edges of the laps the remedy would be to replace the burnt portions of the shells with patches

QUESTION [Rating Boiler Size on Heating Surface]—*In rating boiler horsepower according to the number of square feet of heating surface, is not superheating surface to be included?* L. G.

Answer—The horsepower rating according to heating surface is purely commercial. Water-heating surface, or surface in contact with fire or hot gases on one side and water on the other, is very effective in transmitting heat, and this is

the principal kind of heating surface in nearly all types of boilers and in most boilers it is the only kind. The heat transmission through superheating surface which has fire or hot gases on one side and steam on the other side is very slow, and it is not customary to count in this kind of heating surface in rating the nominal or manufacturer's horsepower of a boiler. The superheating surface should be separately stated as such.

QUESTION [Lower Allowable Pressure Requires Larger Safety Valve]—*Two boilers are of the same dimensions throughout and each is provided with the same size of grate, kind of fuel and draft, and the temperature of feed water in each case is to be* 200 *deg. F. one of the boilers is allowed to carry a working pressure of* 50 *lb. gage and the other* 150 *lb. gage. Which should be provided with the larger safety valve?* E. L.

Answer—The boiler that is allowed the lower pressure should be provided with the larger safety valve. From the identity of the boilers and settings, it is to be presumed that each will transmit the same amount of heat in generation of steam. A pound of steam at 50 lb. gage, or 65 lb. absolute, contains 1178.5 B.t.u., and for generation from feed water at 200 deg. F., each pound would recieve 1178.5 − (200 − 32) = 1010.5 B.t.u. A pound of steam at 150 lb. gage, or 165 lb. absolute, contains 1195 B.t.u., and for conversion of the feed water into steam at this pressure each pound must receive from the boiler 1195 − (200 − 32) = 1027 B.t.u. Therefore, for the same heat generated by the furnace and transmitted by the boiler, there would be $\frac{1010,5}{1027}$, or about 98 per cent. as many pounds of steam generated at the higher pressure. Napier's approximate formula for the discharge of steam from a safety valve is: Flow in pounds per second = absolute pressure × area in square inches ÷ 70; that is, the escape area required is directly as the weight and inversely as the absolute pressure, so that the safety valve for discharging steam at 165 lb. absolute would need to have only 98 per cent. of $\frac{65}{165}$, or about 40 per cent. as much area of opening as for discharging as much steam as would be generated at 65 lb. absolute for the same amount of heat transmitted by the boiler.

QUESTION [Efficiency of Boiler, Furnace and Grate] —*What would be the percentage of efficiency of boiler, furnace and grate obtaining an evaporation equivalent to* 10.95 *lbs. of water from and at* 212 *deg. F. per pound of dry coal containing* 14,645 *B.t.u. per pound?* A. D. K.

Answer—Expressed as a formula, efficiency of boiler, furnace and grate =

$$\frac{\textit{Heat absorbed by boiler per pound of fuel fired}}{\textit{Heat of perfect combustion per pound of fuel}}$$

As evaporation of a pound of water from and at 212 deg. F. requires 970.4 B.t.u., the heat absorbed by evaporation of 10.95 lb. would be $970.4 \times 10.95 = 106,258.8$ B.t.u. and for coal containing 14,645 B.t.u. per pound, the efficiecny of boiler, furnace and grate would be $\frac{106,258.8}{14,645} = 0.7255$, or $72 \frac{55}{100}$ per cent.

QUESTION [Maximum Allowable Working Pressure for Cylindrical Boiler Shell]—*With a factor of safety of* 6, *what would be the highest allowable working pressure for a cylindrical boiler shell* 72 *in. in diameter, made of* ½*-in. steel plate of* 55,000*-lb. tensile strength and* 85 *per cent. efficiency of longitudinal joint?* E. E.

Answer—The formula for determining the maximum allowable working pressure for a cylindrical shell is

$$P = \frac{TS \times t \times E}{R \times FS}$$

in which

P = Maximum allowable working pressure, lb. per sq. in.

TS = Tensile strength of shell plates, lb. per sq. in. cross-sectional area.

t = Minimum thickness of shell plates, in inches.

E = Efficiency of longitudinal joint.

R = Radius = one-half the inside diameter of the outside course of the shell, in inches.

FS = The lowest factor of safety allowed.

Substituting the given values, the formula gives

$$P = \frac{55,000 \times 0.5 \times 0.85}{36 \times 6} = 108.2 \textit{ lb. per sq. in.}$$

maximum allowable working pressure.

QUESTION [Determining Thickness of Bag in Boiler Shell]—*What formula or calculation would be used in computing the thickness of the metal at the center of a bag in a boiler shell?* J. W. M.

Answer—Without making actual measurements inside and outside of the bag, its form and the manner in which the original thickness is reduced are matters of speculation. But if it is assumed that there is a uniform reduction of thickness from the periphery to the center of the bag, the thickness at the center will be approximately twice the original thickness multiplied by the length of the bag taken in a straight line, divided by the curved length, less the original thickness of the shell. It may usually be taken for granted that the thinnest metal will be at the point where the greatest bulge has occurred. The proper method of determining the thickness is to make the measurement with a hook gage inserted through a

hole drilled through the bag. Such a hole would be made for swaging the bag out of the sheet, and, whether or not that is done, a hole thus made afterward may be filled with a rivet.

QUESTION [**Hydrostatic Test Pressure for Boiler**]—*If a boiler safety valve is to be set to blow at 250 lb. gage pressure, what should be the hydrostatic test pressure?* F. S.

Answer—A boiler should be subjected to a hydrostatic test of one and one-half times the maximum allowable working pressure, and the test pressure should be under proper control so that the required test pressure should not be exceeded by more than 6 per cent. If the safety valve is to be set to blow at 250 lb., the boiler should be subjected to a hydrostatic test of $1\frac{1}{2} \times 250 = 375$ lb. per square inch.

QUESTION [**Testing with Air Pressure**]—*Is air pressure as reliable as hydrostatic pressure for testing the tightness of piping, boilers or other pressure containers?* H. T.

Answer—When the test pressure can be held, testing with air pressure is fully as reliable an indication of strength and tightness as testing by hydrostatic pressure. Original testing by air pressure alone is dangerous, for in event of rupture of the pipe or vessel under test, the rapid expansion of high velocity of the escaping air and recoil of the vessel may have the effect of a destructive explosion. For the same reasons serious damage has frequently resulted in making hydrostatic tests of vessels that were partly filled with air. Tests of strength should first be made with hydrostatic pressure 50 per cent. in excess of the allowable working pressure. As air is more searching of leaks, the joints of piping and containers for compressed air or gases may be more thoroughly tested by air pressure, but in cases where it is desirable to insure against an explosive effect, testing by air pressure should be preceded by a higher hydrostatic pressure.

QUESTION [**Water Discharged by Blowing Down $1\frac{1}{2}$ In.**]—*What quantity of water is discharged from blowing a* 72 *in. dia.* × 18 *ft. horizontal return tubular boiler* $1\frac{1}{2}$ *in. from a water level* 17 *in. below the under side of the shell?* P. E. M.

Answer—The cross sectional area of the steam space may be found by the approximate formula for the area of a segment of a circle, namely,

$$\textit{Area of segment} = \frac{4H^2}{3}\sqrt{\frac{2R}{H} - 0.608}$$

where H = the height of segment and R = the radius of arc. When the height of the steam space is 17 in. and $2R$ is equal to 72 in., by substitution in the formula the cross sectional area of the steam space would be

$$\frac{4 \times 17 \times 17}{3}\sqrt{\frac{72}{17} - 0.608} = 733.9 \ \textit{sq.in.}$$

and when the height is $1\frac{1}{2}$ in. lower, or $18\frac{1}{2}$ in. below the sectional area of the steam space would be

$$\frac{4 \times 18.5 \times 18.5}{3}\sqrt{\frac{72}{18.5} - 0.608} = 862.9 \text{ sq. in.}$$

Hence the cross sectional area of the water discharged would be 826.9 − 733.9 = 93 sq.in. and for 18 ft. length of shell, the volume of water discharged would be 93 ÷ 144 × 18 = 11.62 cu.ft.

The weight of water discharged would depend on its temperature. If the boiler pressure was 100 lb. per sq.in., the temperature would be 338.1 deg. F. At that temperature water has a density of 54 lb. per cu.ft. and the weight of water discharged would be about 11.62 × 54 = 627.48 lb.

QUESTION [Material of Clean Tubes Unaffected by Forcing Boiler]—*What is the highest rate of transfer of heat that can be made through water tubes of a boiler without destroying the material?* J. A. G.

Answer—The highest rate of heat transfer has never been known to destroy the metal when the tubes have been clean and kept covered with water. The rate required to burn out the tubes depends on the circulation and the kind and quantity of scale or other coating on the water side of the tubes. When new tubes have been burned out after very short use, the trouble has been traced to presence of oil.

QUESTION [Confining Operations to a Single Boiler]—*We must renew our boiler plant, consisting of two 75-hp. boilers. One of the old boilers is sufficient for our requirements during the summer months, but to supply our winter requirements for steam it is necessary to operate both boilers. Would there be better operating economy to replace the present 75-hp. boilers, or install one 150-hp. boiler for use the year around?* T. B. C.

Answer—By skillful firing and reducing the grate area during the summer season, employment of a single 150-hp. boiler might effect a small saving of fuel, but with only one boiler there would be fewer opportunities for cleaning and making repairs without shutting down the plant.

QUESTION [Measuring Amount of Blowoff]—*What quantity of water will be discharged from a boiler under 160 lb. gage in blowing off to the atmosphere through a $2\frac{1}{2}$-in. blowoff pipe that discharges to the atmosphere?* H. L. S.

Answer—An accurate estimate of flow could not be based on simply the boiler pressure, as the rate of discharge would depend on the pressure obtained in the discharge end of the blow-off line after deducting the losses of pressure in the line. The best method of estimating the amount discharged would be to mark the water level of the boiler and, after blowing off the boiler with all other outlets closed, measure the amount of feed water required to restore the water level.

QUESTION [Order to Cut Boiler Out of the Line]—*When there is a battery of boilers each provided with a superheater and each one connected to a saturated-steam line and a superheated-steam line and an order is given to cut a certain boiler "out of the line," would the order be complied with by only cutting the boiler out of the superheated-steam line?*

J. C. B.

Answer—An order to cut a boiler out of the line is commonly employed and understood to signify cutting the boiler out of common service and connection with other boilers in the same battery; and unless some particular line is specified, the order should be construed as direction for the boiler to be cut off from the saturated-steam line as well as the superheated-steam line.

QUESTION [Laying up Fire-Tube Boilers for a Long Time]—*We have a number of stationary fire-tube boilers of different types that may be laid up out of service for a long time. What is the best method of preventing their deterioration?*

W. L. C.

Answer—When a boiler is laid up out of service, it should be thoroughly cleaned inside and outside and protected from moisture. After thoroughly cleaning the inside, the boiler should be filled nearly full with clean water and then filled to overflowing with a low grade of mineral oil. The water then may be drawn off, leaving all parts of the interior coated with oil. Any water left in depressions then should be driven off with a light wood fire. As an additional precaution against rusting, place a small quantity of unslacked lime in the boiler before closing it up. Thoroughly clean out fire tubes and swab them with oil, paint the outside of flue sheets and plug tube ends with waste. For a locomotive type of boiler, or any but a boiler set in brick work, thoroughly clean the exterior and give it a coat of asphalt or red-lead paint. Boilers that are bricked-in usually are protected from the weather, and their exterior surfaces will not need any further care than to protect them from moisture. Seal fire and ashpit doors tight and seal dampers with asbestos, cotton wool or other convenient filling for crevices and, if practicable, disconnect them from chimneys so that there will be no circulation of air to deposit moisture. To remove oil from the interior of a boiler that has been laid up with an internal coating of oil, fill the boiler nearly full of water, to which add about 10 lb. of soda ash for a boiler rated 50 to 150 hp. and, after boiling for about 10 hours over a slow fire, with 1 to 5 lb. gage pressure, thoroughly wash out the boiler and fittings.

QUESTION [Stopping Leaks at Mud Rings of Old Boilers]—*We have three locomotive-type boilers that leak badly at the mud rings of the water legs. The boilers are badly needed until we have completed the installation of water-tube boilers. I have tried to stop the leaks by calking, but it does not hold.*

Can you suggest an inexpensive method of temporarily stopping the leaks in the old boilers? H. B. B.

Answer—If the openings are small and the water legs of the boilers are washed clean, the leaks probably could be temporarily stopped by spreading bran or cornstarch in the bottom of the water legs to a depth of about one-half inch, the remedy depending on the material becoming deposited in the leaky openings when carried to them by an outflow of water.

QUESTION [Time for Blowing Down Boilers]—*When is the proper time to blow down a boiler, when banked or when steaming at a high rate?* T. B.

Answer—If the water is very muddy, the boiler should be blown down several times a day and preferably during intervals like morning, noon and night, when the boiler is doing the least steaming. If blowing down is done only once a day, the best time is in the morning or after the longest interval of rest and before the settled sediment becomes too much stirred up by the circulation during active steaming.

QUESTION [Obtaining Sample of Flue Gas Analysis]—*At what point of a boiler setting should samples of flue gas be taken for* CO_2 *determination?* J. F.

Answer—Samples of flue gas are usually taken from the breeching between the boiler and stack, and preferably where the gases are just leaving the boiler. The sampling tube should be made of a small size of pipe with a number of $\frac{1}{8}$ in. holes drilled in its sides, and extended nearly all the way across the breeching, so the sample of gas collected will represent the average of flue gases discharged to the stack.

QUESTION [Providing Against Pitting of Boiler]—*What steps should be taken if a boiler plate is found to be pitted?* J. B.

Answer—When pitting is discovered, it should be determined by competent inspection whether the corrosion sufficiently endangers the safety of the boiler to require renewal of the plate, or the highest working pressure that can be carried with safety. When use of the boiler is to be continued, such inspections and determinations should be made at frequent intervals, and to arrest the pitting the feed water should receive suitable treatment for neutralizing the corrosive action of the impurities, as revealed by chemical analysis.

QUESTION [Water-Test Pressure on Handhole Gaskets]—*After washing and resetting handhole covers of a water-tube boiler, is there any objection to testing the tightness of the gasket joints with cold-water pressure pumped up to the running steam pressure for the boiler?* J. R.

Answer—A pumped water pressure, unless made slowly

with a hand pump, is likely to run up in excess of the intended pressure and thereby cause leaks that would not occur with the intended pressure. If gaskets have proper elasticity, and the bearing faces are true, a test-pressure of no more than half of the operating pressure should be sufficient to guarantee tight gasket joints.

QUESTION [**Water Level Equalization Impractical**]—*How can equalizer pipes be arranged to obtain uniformity of water level with a single-feed stop valve in a battery of high-pressure boilers?* C. L.

Answer—Equalizer pipes of reasonable size cannot be depended upon for maintaining a uniformity of water level. This can be appreciated when it is considered that one inch difference of water level would result for a difference of only 1/27 lb.per square inch of pressure of steam generated, or from variation in loss of head of only 1 in. water-column pressure for overcoming the friction of flow in the equalizing connections.

QUESTION [**Equalization of Draft of Boilers Set in Battery**]—*We have three horizontal return-tubular boilers set in battery, each provided with a* 48 *by* 16 *in. uptake, and all discharging into a* 48-*in. diameter stack connection. There is difficulty in obtaining equalization of the draft, especially when one of the boilers is operated with a forced draft. How can the trouble be remedied?* W. G. J.

Answer—For obtaining the most benefit from smoke connections of a given size, all junctions of the uptakes with the main stack connection, and junction of the latter with the stack flue, should be made with easy bends so that flow of the gases may be smooth and continuous. To obtain equalization of the draft, each boiler uptake except the one farthest from the stack should be provided with a baffle for so directing the course of the gases that, when discharged into the main connection, their flow will be toward the stack before coming in contact with gases discharged by other boilers. By this arrangement the gases discharged from a boiler with stronger draft exert a jet-jump action rather than an obstruction to the discharge of the furnace gases from the other boilers.

QUESTION [**Inside and Outside Calking**]—*Is it customary to calk boiler seams inside as well as outside of the boiler?* L. E. F.

Answer—Calking usually is done only on the outside. In marine boilers and very carefully constructed boilers a preliminary calking is given to the inside laps, and absolute tightness is afterward secured by a calking on the outside.

QUESTION [**Regulating Forced Draft of Vertical Boilers**]—*We have three* 150-*hp. vertical fire-tube boilers, operated by forced draft from a fan blower connected by a branch pipe*

with shutoff to each boiler ashpit, and in addition there is a damper in the main pipe from the blower that is controlled by a regulator. Would it not be better to leave these dampers wide open and regulate the draft by means of the uptake damper?

F. A. C.

Answer—The present arrangement is better than regulation of draft by the uptake damper, because any time the latter chanced to be partly or entirely closed, the blower would build up excessive pressure in the furnace, causing hot gases and ashes to be blown out of the fire-doors.

QUESTION [Blowing Off a Cut-Out Boiler]—*When a boiler is cut out from the main line for cleaning, should it be blown out when steam has become reduced to about 15 lb. pressure, or left standing for several days filled with cold water?* A. F. B.

Answer—A boiler should not be blown off under pressure or while the brickwork is hot. Sudden reduction of pressure is injurious to the joints, and if the boiler and brickwork are not cool, the heat will bake the mud and scale in the boiler, making it difficult to clean the shell and tubes. After cutting out, there will be no advantage in allowing the boiler to stand longer than necessary for allowing it and the brickwork to become cooled.

QUESTION [Improvement of Draft by Cleaning Chimney Connection]—*A boiler fireman was informed by his superior that no improvement of draft was to be obtained by cleaning out the smoke connections to the chimney which had become filled about one-third full of soot. What is the explanation of this, if true?* B. F. G.

Answer—For a given force of draft and given size of chimney flue, the volume of furnace gases that can be carried off depends on the least cross-sectional area of smoke-pipe connection and the smoothness of the passages for the gases from the boiler to the chimney. Sharp corners and short bends cause swirls and eddies that give rise to soot deposits which smooth out the channel and usually result in so much higher average velocity that there is a greater volume of gases discharged with the reduced but modified cross-sectional area. Generally speaking, removal of the soot deposits will not improve the volume of draft, unless it results in enlarging the smallest cross-sectional area for passage of the furnace gases. However, no serious error is made by removal of a greater quantity of soot for, in very short time, deposits again form in best arrangement for maximum draft.

QUESTION [Removing Soot from Water-Tube Boiler]—*Is it a good idea to clean soot from the tubes and baffles of a water-tube boiler by washing down with water from a hose?*

R. P.

Answer—Washing with water should be done only when the boiler is cold and will be injurious to the brickwork of

the setting and cause firebrick baffles to crumble if the boiler is used before the setting has been slowly dried out. It is better and also more convenient to remove soot by blowing it off with dry steam, which can be done while the boiler is in use and when there is good draft for carrying off the lightest particles of soot.

QUESTION [Flame in Smoke-Stack of Locomotive Type Boiler]—*What is the cause of smoke burning in the stack of a locomotive-type boiler?* C. E. S.

Answer—In burning solid fuels, the appearance of flame in the stack, or issuing from it, usually is due to particles of burning fuel or heated ash carried by the draft from the furnace and swept through the tubes to be discharged as cinders or sparks. The appearance of combustion is more pronounced when the fire surfaces of the boiler need cleaning of soot, for then the temperature of the gases may remain high enough for the heated particles to continue to glow and present the appearance of burning as discharged from the stack. Flame, with appearance of gases burning at the outlet of a stack, may result from combustion of volatile gases liberated from the fuel by the heat of the furnace, but which have not been mixed with enough air for their combustion until reaching the very point of discharge to the atmosphere.

QUESTION [Feeding Return-Tubular Boiler Through Blowoff]—*What are the advantages and disadvantages of feeding a return tubular boiler through the blowoff?* G. A. B.

Answer—The single advantage of feeding a boiler through the blowoff is that it saves provision of a separate feed connection. The advantage sometimes claimed that, by feeding through the blowoff, the flow of feed water keeps the blowoff flushed of mud, is likely to be defeated by mud filling both blowoff and feed connection when there is intermission of feeding. In cases where the feed water contains impurities, like lime, that are precipitated by heat, feeding through the blowoff is likely to contribute to scaling up of the blowoff connection. Another disadvantage is that by feeding in bulk through a hole in the shell, the greater or less irregularity of discharging feed water at lower temperature than the temperature of the boiler water gives rise to differences of expansion that are injurious to the boiler. All these objections, except scaling and rusting of the feed pipe itself, are overcome by feeding through a pipe introduced through the shell above the fire line and carried thence for some distance through the boiler, so the feed water may be raised more nearly to the boiler temperature and then discharged at some distance from the shell or stays.

QUESTION [More Uniform Pressure from Variable Boiler Pressure]—*We have a direct steam line from our* 100-*hp. return-tubular boiler to a small oven* 60 *ft. above the boiler, for*

which we wish to supply steam at a constant pressure of 5 *lb. gage. The boiler pressure varies from* 70 *to* 90 *lb. The pressure-reducing valve placed near the oven does not operate as closely as we desire. Would it not be better to employ another valve in the boiler room for reducing the pressure to about* 40 *lb.?* W. C. H.

Answer—The trouble is probably due to the wide variation of the initial pressure presented to the reducing valve, and there should be delivery of more uniform pressure from the present reducing valve by placing another reducing valve in the line at the boiler, set to reduce the boiler pressure to an intermediate pressure as proposed.

QUESTION [**Foaming and Priming of Boiler**]—*What is foaming and what is priming of a boiler, and what is the danger of each?* J. K.

Answer—A boiler is said to foam if the steam space is partially filled with unbroken bubbles of steam, and to prime if the steam carries water with it from the boiler. Foaming is caused by any materials either dissolved or suspended in the water which retard or interfere with the free separation of steam from the water. More generally, the trouble is caused by oily or dirty water and can be overcome by feeding and blowing. Where there is a surface blow, it can be used to good advantage. Priming may be caused by foaming or by particles of water being projected into the steam space by the action of boiling or activity of circulation of the water. When due to circulation, it can be reduced and sometimes stopped by checking the fire and carrying lower water level in the boiler. The dangers of foaming and priming are that the boiler-water level may become dangerously lowered from sudden opening of the safety valve or from water carried along with steam supplied for any purpose, and when carried over into an engine, the water is likely to cause a smash.

QUESTION [**Priming of Portable Boiler**]—*What would cause priming of a threshing-engine boiler after about ten days' use with the same quality of fuel and of feed water?* E. R.

Answer—The priming of the boiler may be caused by sediment accumulated out of the feed water, or carrying the water so high as to afford insufficient steam room, or may be due to local violent boiling of the water from greater concentration of the furnace heat upon the heating surfaces of the firebox, caused by dirty fire tubes. Frequent washing out of the boiler and cleaning of the tubes will result in better quality of steam and improvement of fuel economy.

QUESTION [**Air Gathered in Feed-Water Oil Filter**]—*The returns of a vacuum steam-heating apparatus are delivered to an air-separating tank and thence discharged to a receiver and boiler-feed pump. After being discharged by the pump, the return water, on its way to the boiler, is passed through a cloth-*

bag filter for removal of the oil. Air collects in the upper part of the filter. What is the cause and remedy for removal of the air gathered in the filter? G. G. W.

Answer—It is probable that the "air" complained of consists of oil vapor and air liberated out of the water when the pressure of the water is suddenly reduced, after being subjected to the greater pressure necessary for forcing the water through the filter. If it is permissible to have a small air space in the top of the filter, the air or vapor can be relieved automatically by connecting a float type of radiator air valve with the upper part of the filter chamber, or with an appropriate enlargement of that space by employing a small air chamber made of pipe and fittings.

QUESTION [Use of Live-Steam Feed-Water Heater]—*What are the advantages of using a live-steam feed-water heater?* F. R.

Answer—Live-steam feed-water heaters are of advantage when the feed water can be purified by heating it to a high temperature and also in preventing stresses in a boiler due to use of cold feed water. But there is no gain in economy from heating the feed water with live steam, as the heat added to the feed water is taken from the boiler and there is a loss by radiation of heat from the heater and its connections.

QUESTION [Boiler Corrosion from Feeding Heating Returns]—*We are troubled with pitting of the front lower sheet of our horizontal return-tubular boiler and believe it may be due to feeding the condensation returned from an exhaust steam-heating apparatus, as the corrosion does not seem to continue after the heating season is over. How can the trouble be cured?* S. K.

Answer—Corrosion from refeeding condensate might be more rapid because the water is purer, or from free acid due to adulterants in the engine cylinder oil which find their way into the boiler, but in either case all parts of the water space would be affected. If the corrosion is mainly confined to pitting of the fire sheets, they may have been overheated from presence of oil carried over from the exhaust, and once burnt, their corrosion would proceed more rapidly. Better means should be employed for removing oil from the exhaust and condensate before refeeding. Use engine cylinder oil sparingly, wash the boiler more frequently and if the boiler has to be forced, do not feed the return water from the exhaust-steam heating system.

QUESTION [Gain from Heating Feed Water]—*For generation of steam at 90 lb. gage, what is the percentage gain when boiler-feed water is heated from 50 deg F. to 210 deg. F.?* R. L.

Answer—A pound of dry saturated steam at 90 lb. gage, or 105 lb. absolute, contains 1187.0 B.t.u. above 32 deg. F., and when the feed water is at the temperature of 50 deg. F., or $(50 - 32) = 18$ deg. above 32 deg F., for conversion into steam at the stated pressure each pound must receive $1187.2 - 18 = 1169.2$ B.t.u. When the feed water is at the temperature of 210 deg. F., each pound of feed water evaporated requires $210 - 50 = 160$ B.t.u. less, or only $(1169.2 - 160) \div 1169.2 \times 100 = 86.3$ per cent. as much heat for conversion into steam, and the gain would be 13.7 per cent.

QUESTION [**Saving from Higher Temperature of Feed Water**]—*Where the boiler pressure is 120 lb. gage, what is the percentage of saving from returning the condensation from heating coils to the boiler at 240 deg. by traps in place of returning the water to a tank and feeding the water at 180 deg. F.*

P. E. M.

Answer—In either case each pound of feed water, when converted to steam at 120 lb. gage, or 135 lb. per sq. in. absolute, would contain 1191.6 B.t.u. above 32 deg. F. When the feed temperature is 180 deg. F., then each pound of feed water for conversion into steam would have to receive $(1191.6 + 32) - 180 = 1043.6$ B.t.u.; and when the feed temperature is 240 deg. F, each pound of the feed water would have to receive $(1191.6 + 32) - 240 = 983.6$ B.t..u, or $(1043.6 - 983.6) \times 100 \div 1043.6 = 5.75$ per cent. more heat. Whether this represents the actual saving of coal from use of the traps, in place of a receiving tank and feed pump, depends on the relative steam economies of the trap cr feed pump that may be employed.

QUESTION [**Equivalent Evaporation at Different Feed-Water Temperatures**]—*How many pounds of feed water at 212 deg. F. evaporated into steam at 70-lb. gage pressure would be equivalent to the evaporation of 30 lb. of water from a temperature cf 100 deg. F. into steam at the stated pressure?*

J. R. C.

Answer—A pound of steam at 70-lb. gage pressure, or 85 lb. absolute, contains 1183.4 B.t.u. above 32 deg. F., and evaporation of 30 lb. of feed water from a temperature of 100 deg. F into steam at 70-lb. gage pressure would require $[1183.4 - (100 - 32)] \times 30 = 33{,}462$ B.t.u. For evaporation from a temperature of 212 deg. F. into steam at 70-lb. gage each pound of feed water would require $1183.4 - (212 - 32) = 1003.4$ B.t.u., and to obtain an evaporation into steam at 70-lb. gage pressure equivalent to conversion of 30 lb. of feed water from a temperature of 100 deg. into steam at 70-lb. gage pressure would require the evaporation of $33{,}462 \div 1003.4 = 33.35$ lb. of the feed water at the temperature of 212 deg. F.

QUESTION [**Permissible Back Pressure for Heating Feed Water**]—*To what extent is it economical to increase the*

back pressure of a non-condensing engine for raising the temperature of boiler-feed water delivered by an exhaust-steam feea water heater? A. R. D.

Answer—Within moderate limits, the total steam consumption of an engine increases substantially in direct proportion to the increase of the load, or increase of mean effective pressure, and each additional pound of average back pressure increases the total steam required for a given load the same as though the load had been increased to require as much additional mean effective pressure as the added back pressure. Under average conditions, where an exhause-steam feed-water heater is employed only for heating water for generation of steam supplied to a non-condensing engine, for one pound of additional back pressure on the engine there will be obtained an increase of about 3 degrees additional temperature of feed water, depending on the efficiency of the exhaust-steam feed-water heater, and for each degree of feed water there will be a saving of about 1/10 of one per cent of heat required for generation of the steam used by the engine. Under these conditions, the limit of advantage is reached when the cost per degree increase of the feed water is ⅓ lb. per sq. in. of additional back pressure, and when for each degree rise of feed-water temperature the increase of back pressure amounts to 1/10 of one per cent of the mean effective pressure.

QUESTION **[Separate Cold-Water Feed Connection for Boiler]**—*What is the advantage of supplying a boiler with a separate cold-water feed line?* R. St. T.

Answer—A separate cold-water feed line is desirable for filling the boiler after cleaning and for testing purposes, and also for regular feeding during such emergencies as when the feed-water heater lines are inoperative from corrosion or incrustation with scale, or during temporary disuse of the heater.

QUESTION **[Designations of Boiler-Feed Waters]**—*What proportion of scale-forming substances is allowable for good boiler-feed water?* R. E.

Answer—The term "good" as applied to feed waters is only relative. The following designations are generally used, based on the number of grains of scale-forming ingredients per gallon of the feed water:

Less than 8 gr. per gal.—Very good.
From 8 to 12 gr. per gal.—Good.
From 12 to 15 gr. per gal.—Fair.
From 15 to 20 gr. per gal.—Poor.

When there is over 20 gr. of scale-forming materials to the gallon, the water should not be used unless first purified.

QUESTION **[Inspecting Leakage of Closed Feed-Water Heater]**—*How can it be known whether a closed feed-water heater is leaking?* J. N.

Answer—Leakage of a closed exhaust-steam feed-water heater will be revealed by a discharge of water along with the exhaust, especially from the drips. The best test of leakage is to observe whether there is discharge of water from the drips when the feed pump is operated without the engine or other exhause discharging through the heater.

QUESTION [**Saving from Use of Feed-Water Heater**]—*What is the percentage of saving by the use of an exhaust steam feed-water heater where the boiler pressure is 90 lb. gage and the feed water is heated from 50 deg. to 170 deg. F.?* R. A. B.

Answer—By referring to steam tables it may be seen that a pound of dry saturated steam at the pressure of 90 lb. gage, or $90 + 15 = 105$ lb. per sq. in. absolute, contains 1,187.2 B.t.u. above 32 deg. F. When the feed water is at 50 deg. F., or $50 - 32 = 18$ deg. above 32 deg. F., for conversion into steam, each pound of the feed water requires $1{,}187.2 - 18 = 1{,}169.2$ B.t.u. If the feed water has a temperature of 170 deg. F., then for its conversion into steam each pound would require $170 - 50 = 120$ B.t.u. less than when the temperature is 50 deg. F., and if the feed-water heater is operated without increasing back pressure on the engine, the saving would be $120 \div 1{,}169.2 = 0.1026$ or about $10\frac{1}{4}$ per cent.

QUESTION [**Corrosion of Feed-Water Heater**]—*What would cause rapid pitting of the shell of a closed exhaust-steam feed-water heater?* L. H.

Answer—Galvanic action is likely to take place with some corrosion of the shell when the heater is provided with tubes and shell of different metals, and corrosion of the kind is intensified by either an acid or an alkaline feed water when the feed water is in contact with the shell. In any form of heater corrosion may be caused by acid feed water or induced by electrolysis due to stray electrical currents.

QUESTION [**Testing Feed Water for Oil Received in Open Heater**]—*How can a test be made to determine the amount of oil added to the feed water by an open exhaust-steam feed-water heater?* F. P. H.

Answer—Make up about one pint of a concentrated solution of caustic soda and dilute with an equal quantity of water, using melted snow or ice water or the purest water conveniently obtainable. Provide two clean glass beakers and fill one about half full with a sample of the water received by the heater; mark the height on the side of the beaker and turn the water over into the other beaker. Refill the first beaker to the same point with a sample of the water discharged from the heater, and beginning with about 4 oz. of the caustic soda solution in a measuring beaker such as used by chemists and photographers, add sufficient of the soda solution to the sample of discharge water to cause it to become decidedly turbid,

then add precisely the same amount of the soda solution to the sample of the water received by the heater. If the latter becomes turbid, repeat the experiment with perfectly clean beakers, each time using smaller equal quantities of the soda solution. This done, each beaker will contain the same quantity of soda solution and of the respective samples of water, and it may be noted how much engine-cylinder oil needs to be added to the beaker containing the sample of water received by the heater to bring it to the same appearance of turbidity as the contents of the beaker containing the same amount of water discharged by the heater and the same dose of soda water. This will not only indicate the presence of oil but, by repeating the experiment on a large enough scale, will demonstrate the ratio of the quantity of oil to quantity of sample water tested.

QUESTION [**Causes of Failure of Injector**]—*What may be the cause of failure in operation of an injector? The tubes and parts are clean and show very little wear.* W. H. A.

Answer—The trouble may be due to a leaky suction pipe. Plug the ends of the suction and overflow pipes. Then turn on the steam supply and the location of any leaks will be indicated by an escape of steam. The injector may be too hot to start, due to a leaking steam valve, or the feed water may be too hot. The suction pipe or strainer may be obstructed. Such obstructions usually can be removed by turning on steam after blocking the overflow and closing the delivery to the boiler. The delivery pipe may be choked or the check-valve may be faulty, either of which would be indicated by discharge at the overflow or inability to maintain the usual water level in the boiler. To test for these defects, place an accurate pressure gage on the delivery side of the injector and note whether it indicates pressure in excess of the boiler pressure It may be that the steam supply pipe is choked or that the injector is supplied by wet steam, caused by foaming or priming of the boiler. The steam supplied should be taken from an independent connection out of the highest part of the boiler to insure dryness. On the other hand, superheated steam will not usually give results as satisfactory as dry saturated steam, as sufficient time may not be permitted for its condensation in the injector.

QUESTION [**Testing Surface Condenser for Leakage**] —*How may a surface condenser be tested for leakage?* W. B. R.

Answer—Probably the best method to determine the amount of condenser leakage is to pass cooling water through the tubes at the normal rate of flow, at the same time maintaining the required vacuum with the air pump. The amount of water removed by the air pump is the leakage of circulating water into the steam space of the condenser. The most general test for determining whether or not there is any leakage

is to close all connections and observe whether a vacuum once established can be maintained a reasonably long time. Where salt water is used for cooling, it can be determined while running whether there is any leakage, by adding a little silver nitrate to a small amount of the condensed steam; if there is no appreciable precipitate of white silver chloride, it may be assumed that the condenser does not leak.

QUESTION [Connecting Live-Steam Heating to a Condenser]—*Is there any benefit in connecting a condenser on heating coils supplied by high-pressure steam direct from the boiler except for the purpose of collecting the condensation for return to the boiler?* S. C.

Answer—Any form of condenser for continuous operation would be provided with means for discharge of air, and that feature of operation would be beneficial to any system of steam heating to prevent formation of air pockets and thereby increase activity of the circulation of the steam. The heat that is imparted to the condenser is lost if the condensing water is discharged as waste or is of no benefit in increasing the temperature of the boiler-feed water. But for most purposes of live-steam heating, and especially where the steam is used at reduced pressure, a condenser and air-pump connection is of benefit for at least occasional removal of air without extravagant waste of steam, as by blowing out returns and leaving air vents open to the atmosphere. It should be borne in mind, however, that condensation is necessarily accompanied by a drop in pressure and temperature in the heating system, and during operation of the condenser it deprives the heating system of steam that must be made up in the supply, and the heat thus absorbed by the condenser is lost unless utilized.

QUESTION [Condenser Pressure Not Ascertainable from Temperature]—*Why cannot condenser pressure be known by simply obtaining the temperature and ascertaining the corresponding pressure from tables of the properties of steam?* W. A.

Answer—The absolute pressure within a condenser is made up of the pressure due to the steam or water vapor present and the pressure due to the gases present, generally called air as the greater part of such gases usually consists of air. The total condenser pressure cannot be obtained from steam tables because the temperature is that corresponding to the pressure of only the saturated steam that is present, which would be something less than the total pressure, depending on the amount of "air" contained in the mixture.

QUESTION [Draft Affected by Direction of Wind]—*Does the direction of the wind have any influence on the way boilers steam? We have four boilers connected to a* 110-*ft. stack. Our boilers face the south, and when the wind is from the north*

it is harder to hold the steam. There are no hills or valleys to change the wind from its course? M. C.

Answer—The general trend and pressure of the atmosphere are greatest in direction of the wind and, other conditions being equal, there is somewhat lower supply of air and consequently a lower rate of combustion and generation of steam when the direction of the wind is away from the front of the boilers.

QUESTION [Designations of Pressure]—*What is the difference between absolute pressure, boiler pressure and gage pressure?* J. B. S.

Answer—Absolute pressure is the total pressure above a perfect vacuum. Boiler pressure is the pressure resisted by the strength of the boiler, hence boiler pressure is absolute pressure less the exterior pressure of the atmosphere, or, as commonly expressed, is the pressure above atmosphere. The pressure of the atmosphere acts on the exterior of a gage tube or a diaphragm just as it acts on the shell of a boiler. Ordinary pressure and vacuum gages indicate 0 for the same pressure on both sides of tube or diaphragm, and the dial is laid off to indicate pressures above or below the pressure of the atmosphere, hence "gage pressure" means the same thing as "boiler pressure."

QUESTION [Saturated and Superheated Steam]—*What is meant by saturated and superheated steam?*
H. M. C.

Answer—Steam that is at the evaporation temperature corresponding to its pressure is said to be "saturated steam." When steam formed in a boiler rises from the surface of the water, it is saturated, and will remain so as long as it is in contact with the water, and so long as the steam is in a boiler in close communication with the water it cannot be other than saturated, for any attempt to heat the steam to a higher temperature will fail, as the heat will be transmitted to the water and cause further evaporation. Saturated steam may be perfectly dry, or it may be "wet" saturated steam containing suspended particles of water, just as water is suspended in the atmosphere during a fog. Whenever steam is mentioned without qualification, it is understood to be dry saturated steam. Superheated steam is steam whose temperature is higher than the evaporation temperature corresponding with its pressure, and to effect superheating of saturated steam it must be heated away from water to a higher temperature than the temperature of saturated steam of the same pressure and without changing the pressure.

QUESTION [Heat Required for Generation of Wet Steam]—*What is the formula for determining how much heat is required for generation of a pound of steam at* 150 *lb. gage*

and quality 98 per cent. from feed water at the temperature of 120 deg. F.? M. D.

Answer—The number of B.t.u. contained per pound of the steam is given by the formula,

$$H_w = h + qL, \qquad (1)$$

in which

H_w = Number of B.t.u. above 32 deg. F. in a pound of the wet steam;

h = Heat of the liquid, or number of B.t.u. required to heat a pound of water from 32 deg. F. to the temperature of evaporation corresponding with the pressure;

L = Latent heat of evaporation, or number of B.t.u. contained by a pound of dry saturated steam at the stated pressure, in addition to h;

q = Quality or fraction of a whole pound of the steam that has received the latent heat of evaporation.

If t = the temperature of the feed water, than $t - 32$ = the number of B.t.u. contained by the water above 32 deg. F. before any heat is added to it in the boiler; and if x is the quantity of heat required for generation of a pound of the steam, then

$$x = H_w - (t - 32) = H_w + 32 - t \qquad (2)$$

and by substituting the value of H_w in (1),

$$x = h + qL + 32 - t \qquad (3)$$

For steam at 150 lb. gage, or 165 lb. absolute, the tables of properties of dry saturated steam give h = 338.2 and L = 856.8. With q = 98 per cent. and t = 120, by substituting in (3) the heat required is found to be

$$x = 338.2 + (0.98 \times 856.8) + 32 - 120 = 1089.8\, B.t.u.$$

QUESTION [Meaning of "Pound of Steam"]—*What is meant by a pound of steam? Is it not a pound of water evaporated into a pound of steam?* W. D.

Answer—The expression "a pound of steam" means mass of the substance whose weight is one pound avoirdupois, just as signified by the terms a "pound of feathers" or a "pound of lead." When a pound of water is evaporated, it is converted into a pound of steam, and when the steam is condensed it is converted back to the same weight of water; consequently, the weight of steam is commonly expressed in pounds weight of water required for generation of the steam, or pounds of water resulting from its condensation. When speaking with reference to the condition of the steam, the expression "pounds of steam," as most commonly used, refers to the pressure exerted per square inch above the atmosphere, or "gage pressure," but when quantity is referred to, the expression "pounds of steam" refers to the weight of steam under consideration.

QUESTION [**Approximate Formula for Total Heat of Steam**]—*What is the formula for obtaining the total heat of a pound of dry saturated steam, knowing the temperature?*
R. L.

Answer—The total heat of steam, or number of B.t.u. required to raise a pound of water at 32 deg. F. to a given temperature and convert it all into dry saturated steam at that temperature, may be approximately calculated for a temperature t of the Fahrenheit scale by the formula,

$$\textit{Total Heat} = 1086 + 0.305t.$$

On account of the variation of the specific heat and heat transformed into work in raising the water to different temperatures and converting it into steam at corresponding pressures, a simple and precise equation that will cover all temperatures cannot be formulated. For application to ordinary temperatures of steam in rough calculations, the formula quoted may be considered sufficiently accurate, but for close computations the total heat values employed should be obtained by consulting standard tables of the properties of steam.

QUESTION [**Heat of Water and of Steam**]—*Does a pound of water contain as much heat as a pound of steam in the same boiler?*
R. R.

Answer—The water and the steam in presence of the water are of the same temperature, but a pound of the water does not contain as many heat units as a pound of the steam, because conversion of the water into steam requires addition of the latent heat of evaporation.

QUESTION [**Advantages of Early Release**]—*What is the advantage of releasing steam before the end of the expansion stroke?*
W. B. G.

Answer—Since some area of the indicator diagram must be lost in expelling the exhaust it is better that the loss should be above than below the expansion line, for in bringing the piston to rest at the end of the stroke there is no object in keeping up the forward pressure near the end of the stroke at the expense of requiring higher compression on the opposite side of the piston. When a condenser is used, release before the end of the stroke insures prompter realization of the vacuum for the return stroke with a larger percentage of the load thrown on the condenser, as the larger area in the vacuum portion of the diagram calls for earlier cutoff and less initial steam.

QUESTION [**Additional Steam for Increased Back Pressure**]—*How does increase of back pressure affect the quantity of steam required by an engine?*
J. R. A.

Answer—For the same load on the engine each pound of additional average back pressure will require an additional pound of average forward pressure, so that to meet an increase

of back pressure, the supply of steam will have to be increased as much as though the load had been increased to require the same additional mean effective pressure.

QUESTION [Air Contained in Steam]—*Does steam contain the same amount of air as the water from which it is generated, or is there a different amount present?* J. H. B.

Answer—When water is heated to the boiling point, practically all of the air becomes liberated and that weight of air is present as a mechanical mixture with the steam at the same pressure. When the steam is condensed to water, nearly all of the air that was mixed with the steam becomes disengaged as air saturated with moisture, the amount of moisture depending on the temperature at which condensation takes place.

QUESTION ["High-Pressure" and "Low-Pressure" Steam]—*What is the difference between high- and low pressure steam?* L. B.

Answer—The higher the pressure of steam the greater its density, the higher its temperature and the greater the number of heat units contained per pound of heaviness. In other respects the physical characteristics of different pressures are identical. The designations "high-pressure" and "low pressure" steam are purely empirical and vary in significance in different localities, extablishments and operations. High-pressure steam generally is understood to apply to steam received directly from a boiler at high enough pressure for power purposes and without having sustained a reduction of pressure by passage through an engine, pressure-reducing valve or heating apparatus; and "low-pressure" steam generally is applied to exhaust steam or to steam that has been used first-handed in an engine, or steam whose pressure is not high enough for use in a non-condensing engine, or whose pressure has been lowered by passage through a reducing valve or in the performance of some useful operation.

QUESTION [Drier Steam With Reduction of Pressure From Wiredrawing]—*How does steam become drier or superheated by the process of wiredrawing?* E. M.

Answer—When steam flows through an aperture the discharge takes place in consequence of the difference in pressure on opposite sides of the aperture, and in its passage through a reducing valve or other form of opening, without doing any external work, there is practically neither loss nor gain of heat per pound of the steam. There is relatively small loss due to work required for overcoming inertia, and the heat that is converted into work required for overcoming friction is restored to the steam or, under favorable conditions, a portion of it may be lost by radiation. Hence, ordinarily, after thus expanding to a reduced pressure, each pound of the steam contains more heat than necessary for the same

percentage of dryness that it had at the initial pressure; so that the process of wiredrawing, whether performed by passing the steam through a throttled valve, a pressure-reducing valve or other form of aperture, converts wet steam into a drier steam at the lower pressure, dry steam to steam that is superheated for the lower pressure, or may convert superheated steam to steam at a lower pressure with more degress of superheating for the lower pressure than it had for the initial pressure.

QUESTION [Factor of Evaporation Generating Superheated Steam]—*What is the factor of evaporation when, under actual conditions, a boiler generates steam at* 200 *lb. gage with* 100 *deg. superheat from feed water at* 212 *deg. F.; and how many pounds of water have to be evaporated per boiler horse-power?* J. J. R.

Answer—An absolute pressure of 200 lb. gage would be equal to 215 lb. per sq. in. absolute, and, according to the steam tables, a pound of steam at that pressure superheated 100 deg. F., contains 1259 B.t.u. above 32 deg. F., so that if the feed water temperature is 212 deg. F., each pound of feed water for conversion into steam of the stated pressure and quality must receive $1259 + 32 - 212 = 1079$ B.t.u. Unless otherwise provided, a combined boiler and superheater should be treated as one unit and the equivalent of the work done by the superheater should be included in the evaporative work of the boiler. Evaporation of a pound of water from and at 212 deg. F. requires 970.4 B.t.u. and therefore, under the actual conditions, the factor of evaporation would be $1079 \div 970.4 = 1.111$; and as a boiler horsepower is equivalent to the evaporation of 34.5 lb. of water from and at 212 deg. F. per hour, under the conditions stated a boiler horsepower would require evaporation of $34.50 \div 1.111 = 31.05$ lb. of water per hour.

QUESTION [Relative Heat in Water and Steam of Boiler]—*Does a pound of water in a boiler contain as many heat units as that quantity converted into steam at* 100 *lb. or any other pressure?* R. B.

Answer—The water in a boiler is at the same temperature as the steam but before the water can be converted into steam it must receive additional heat, called the latent heat of evaporation. Thus for 100 lb. gage pressure the water and steam are at a temperature of 338 deg. F. and each pound of the water contains 309 B.t.u. above 32 deg. F., but for conversion into steam, each pound of water must, in addition receive 879.8 B.t.u. which is the latent heat of evaporation at the stated pressure; so that the total heat contained by a pound of the steam is $309 + 879.8 = 1188\ 8$ B.t.u.

QUESTION [Transfer of Heat Requires Difference of Temperatures—*If at sea level fifteen pounds of steam at at-*

mospheric pressure is to let into an open barrel containing 150 *pounds of water at the boiling point, what temperature would the water attain?* C. E. G.

Answer—The temperature of dry saturated steam at atmospheric pressure is 212 deg. F., and the boiling point of water under atmospheric pressure is at the same temperature. There could be no transfer of heat because the temperature would be the same and the temperature of the water would remain unchanged.

QUESTION [Pressure Values Given in Steam Tables]—*Why are not steam tables arranged to show temperatures and total heat of steam for each pound of gage pressure instead of absolute pressures?* W. D. P.

Answer—The temperature and heat per pound (weight) of dry saturated steam depends on the absolute pressure, while gage pressure is the pressure above atmospheric pressure, which is variable with the elevation of the place and also varies with changes of atmospheirc conditions. Therefore steam tables based on gage pressure would need to be adapted to a particular pressure of the atmosphere, and greater convenience in their use would be confined to cases where the atmospheric pressure was the same as that for which the tables were particularly constructed.

QUESTION [Pressure Available from Steam Line]—*A* 2-*in. steam line supplies steam to a dry room from a boiler that carries steam at* 100 *lb. pressure and the line is extended beyond a dry room branch to supply steam to three pumps. Is it possible for the pumps to be supplied with steam at full pressure or for the pump at the end of the line to obtain as much steam as the others?* J. M.

Answer—When any steam is drawn from or condensed in a pipe line the line pressure cannot be as high as boiler pressure, because drop in pressure is a necessary accompaniment of flow or condensation. It is therefore impossible for the pumps to be supplied with steam at the full boiler pressure and the greater the draft of steam between the boiler and a branch out of the line the lower the pressure in the line at the point where that branch is taken off. When steam is supplied to any other branch, the line pressure that is available for the branch to the pump at the end of the line cannot be as great as the line pressure available to a pump or any other branch that is taken out of the line nearer the boiler.

QUESTION [Superheated Condition for Reduced Pressure]—*On a steam line that carries dry saturated steam at* 100 *lb. gage pressure, I have placed a reducing valve to reduce the pressure to* 20 *lb. Will the steam delivered to the low-pressure side be superheated?* E. G. G.

Answer—Neglecting the heat lost by radiation, a given weight of steam discharged through the reducing valve will

contain as much heat as it had before passing through the valve. A pound of dry saturated steam at the pressure of 100 lb. gage, or 115 lb. per sq. in. absolute, contains 1,188.8 B.t.u., and a pound of dry saturated steam at the pressure of 20 lb. gage, or 35 lb. per sq. in. absolute, contains 1,166.8 B.t.u. Hence, without loss of heat in passing through the reducing valve each pound would contain 1,188.8 − 1,166.8 = 22 B.t.u. more than necessary for a saturated condition at the reduced pressure, and therefore the steam would be in a superheated condition for the reduced pressure. Referring to tables of the properties of superheated steam, it may be seen that for a pressure of 20 lb. gage, or 35 lb. per sq. in. absolute, and a heat content of 1,188.8 B.t.u., the temperature would be 303.7 deg. F., and as the temperature of dry saturated steam at the pressure of 20 lb. gage is 259.3 deg. F., the steam would be superheated 303.7 − 259.3 = 44.4 deg. F.

QUESTION [Temperature for Mixture of Steam and Water]—*If five pounds of dry steam at a pressure of* 5 *lb. gage are mixed at atmospheric pressure with* 10 *lb. of water at* 60 *deg. F., what would be the temperature and condition of the resulting mixture?* F. B.

Answer—One pound (weight) of dry saturated steam at the pressure of 5 lb. gage or 20 lb. absolute contains 1,156.2 B.t.u. above 32 deg. F., and five pounds of such steam would contain 1,156.2 × 5 = 5,781 B.t.u. above 32 deg. F. A pound of water at 60 deg. F. contains 60 − 32 = 28 B.t.u. above 32 deg. F. and 10 pounds would contain 28 × 10 = 280 B.t.u. above 32 deg. F. Mixing the steam and water together would make a total of 5,781 + 280 = 6,061 B.t.u. above 32 deg. F., and a total weight of 5 lb. steam + 10 lb. water = 15 lb., and each pound of the mixture would contain 6,061 ÷ 15 = 40.4 B.t.u. above 32 deg., so that the temperature of the mixture would be 40.4 + 32 = 72.4 deg. F.

QUESTION [Discharge of Steam Jets]—*What quantity of steam at* 100 *lb. boiler pressure would be used by a steam jet having four nozzles each* $\frac{1}{16}$ *in. dia. discharging into a boiler ash pit?* C. H. T.

Answer—The discharge would be given approximately by Napier's rule for flow of steam through orifices when the final absolute pressure is less than 58 per cent of the initial absolute pressure, namely,

Flow in pounds per second = absolute initial pressure × area orifice in sq. in. ÷ 70

Having an initial absolute pressure of 115 lb. per sq. in. and 4 orifices, each $\frac{1}{16}$ in. dia. their total area would be $\frac{1}{16} \times \frac{1}{16} \times 0.7854 \times 4 = 0.00307 \times 4 = 0.01223$ sq. in., and by the formula the discharge would amount to 115 × 0.01228 ÷ 70 = 0.02017 lb. of steam per second, or 0.02017 × 60 × 60 = 72.6 lb. per hour.

QUESTION [Live Steam Wasted Through Back-Pressure Valve]—*Why is our boiler called upon to generate more steam when heating the building by adding live steam to the exhaust of the engine operated with a light load than when there is a heavy load on the engine and plenty of exhaust for the heating without addition of live steam?* C. T. B.

Answer—The greater amount of steam used when the exhaust is supplemented by live steam undoubtedly arises from the escape of more steam to the atmosphere through the exhaust back-pressure relief valve. When live steam is used to supplement exhaust for heating, it should be supplied through a pressure-reducing valve that will regulate the supply at a pressure several pounds below the relief pressure of the back-pressure valve. Otherwise there is likely to be great waste of live steam in attempting to satisfy the requirements for heating.

QUESTION [Slower Cooking in Trapped Steam-Jacketed Kettles]—*Why should the same amount of cooking in our steam-jacketed kettles require about twice as much time when the discharge of condensate is controlled by traps as when the kettles are supplied with steam of the same pressure and the condensate is discharged by an open bypass?* P. E. N.

Answer—When the traps are used there is less active circulation of the steam and slower absorption of heat by the heating surfaces of the kettles, mainly on account of air pockets formed in the steam spaces of the kettles, which are more thoroughly swept out when the flow of steam is continuous.

QUESTION [Utilization of Discharge from Steam Kettles]—*How can we utilize the heat of exhaust from steam kettles? Our engine exhaust is insufficient for warming our buildings. Can we not combine the exhaust from the steam kettles with the engine exhaust?* T. A. M.

Answer—When properly operated, the discharge from steam kettles should consist of only hot water formed by condensation of the steam used, and the most economical utilization of the heat remaining in the water would be to deliver it by a return trap to the boiler.

QUESTION [Temperatures Within Boiler Water Column]—*In a water column of a boiler carrying steam at a gage pressure of* 125 *lb. per sq. in., what would be the temperature of steam in the water column* $\frac{3}{4}$ *in. above the level of the water and the temperature of the water* $\frac{3}{4}$ *in. below the surface covered by steam?* J. V. H.

Answer—For all practical purposes the temperature $\frac{3}{4}$ in. above or below the surface of the water would be the same as that of dry saturated steam at a pressure of 125-lb. gage; namely, 353 deg. F. Radiation of heat usually is taking place from the steam connection of water columns of boilers while under steam pressure, and consequently there is slightly less

pressure and lower temperature of the steam at or near the water level in the column and gage-glass than the temperature of steam and of water in the boiler; but ordinarily, the difference would be imperceptible. However, the column might be so exposed to cold that the water any distance below the surface would be considerably lower in temperature than the temperature of the steam, depending on circulation and insulation. The temperatures are controlled by the same principles as those which determine temperatures within pipes and radiators of a two-pipe gravity-return steam-heating system. The steam connection of the water column corresponds to the supply line, the steam space corresponds to the radiators and portions of the system not flooded with water, and the water connection of the water column corresponds to the flooded return pipes of the heating system, which may become sufficiently chilled by the surrounding atmosphere for the water within them to become frozen from the radiators clear back to the boiler.

QUESTION [Taking Saturated Steam from Boilers Having Superheaters]—*All our boilers are equipped with superheaters, but one of our engines does not work well with superheated steam. How can this engine be supplied from the boilers?* J. G. S.

Answer—For supplying the engine with saturated steam from the boilers, the steam would have to be taken out of the boilers by separate saturated-steam outlets, or might be taken out of the boiler connections to the superheaters between the saturated-steam space and the superheaters.

QUESTION [Advantage of Superheated Steam for Reciprocating Engine]—*How is advantage gained by the use of superheated steam for a reciprocating engine?* G. L.

Answer—The principal advantage is derived from prevention of initial condensation. The higher the degree of superheat the drier the steam at cutoff, and by sufficiently superheating the steam supplied to an engine, it is possible to prevent any cylinder condensation and deliver dry or superheated steam during the exhaust. By using superheated steam, single-cylinder engines which are subject to large losses from cylinder condensation when supplied with saturated steam, may be operated successfully through a greater range of expansions and such an engine may be run economically with either a light or a heavy load.

QUESTION [Reporting Average Cost per Pound of Steam]—*The monthly report blanks of our vacuum-heating plant provide for "average cost per pound of steam in system" and "average cost per pound of steam in boiler." How should these items be computed?* F. D. C.

Answer—For obtaining the "average cost per pound of steam in the system," divide the total cost of fuel, labor and

other expenses for the month, by the total number of pounds of steam supplied to the coils or radiators, as ascertained by weighing or metering the water that has been discharged by the vacuum pump, less the weight of injected water. For ascertaining the "average cost per pound of steam in boiler," divide the total cost by the number of pounds of water fed to the boiler.

QUESTION [Advantages and Disadvantages of Steam Dome]—*What are the advantages and disadvantages of providing a horizontal return-tubular boiler with a steam dome?*
L. D.

Answer—The advantage of a steam dome on a boiler is that it increases the volume of the steam space and allows the steam to be taken from the boiler at a point somewhat removed from the surface of the water, thereby insuring a supply of drier steam than if the supply were taken directly from the shell. The leading disadvantages are the added expense, difficulty of making and maintaining safe and tight connections of the dome with the shell and the uncertain weakening effect of the dome opening on the shell. The advantage of obtaining drier steam can be met, however, by a good form of dry pipe or separator placed within the shell or by employment of an exterior steam drum. A steam drum with nozzle connection to the shell is usually less expensive than a dome, and besides affording the opportunity for much safer construction has all the advantages without any of the disadvantages of a dome, excepting the requirement of practically the same if not less headroom.

QUESTION [Greater Efficiency of Exhaust Steam for Heating]—*What is the reason for exhaust steam heating up a building quicker than live steam?* W. B.

Answer—Exhaust steam usually is wetter, containing more water swept along with the steam at the same temperature than in case of steam of quality generally discharged by a low-pressure heating boiler, or that is made drier when discharged through a pressure-reducing valve. Exhaust steam thereby contains rather more heat units per cubic foot, and the conduction of heat to pipes, coils and radiators is more rapid with wet steam than with dry steam. But where the steam supplied is of the same actual quality and pressure, there is no difference in the time required for heating up a building by live or exhaust steam, and the greater efficiency commonly attributed to exhaust steam is generally due to a more liberal supply and maintenance of actually higher pressure and temperature of the steam.

QUESTION [Circulating Exhaust Steam in Vertical Loop Radiators]—*For drying grain we have four vertical-pipe radiators supplied with exhaust steam, from which we do not obtain sufficient heat for our purposes. Each radiator contains*

800 ft. of 1-in. pipe screwed into a cast-iron base. The pipes are in pairs having their upper ends connected with return-bend fittings. Could we not get better results if we connected each radiator drain with an open-air vent pipe? C. B.

Answer—On account of the difficulty of removing air from the loops, good circulation of exhaust steam is not readily obtained in this type of radiator, especially if the pressure of the steam as supplied to the radiator is only a pound or two above the pressure of the atmosphere and the returns are discharged direct to the atmosphere. Hence, operation of the radiators would be improved but little if at all by independent vents to the atmosphere, and there would be much waste of steam by short-circuit through the bases of the radiators. Better results would be obtained by connecting the returns to a vacuum system capable of maintaining at least 10 in. vacuum, for promptness in removing the air when starting up the system, after which the vacuum could be reduced to the requirement for maintaining satisfactory circulation.

QUESTION [Pressure for Operating Heating Boiler]—*The boiler used for heating our building is operated at 85 lb. pressure, and the steam is supplied through a reducing valve at 10 lb. pressure. Would it not be more economical to operate the boiler at 10 lb. pressure and supply the steam direct.*

C. P. R.

Answer—There is neither loss nor gain of heat from passing steam through a reducing valve, excepting the small loss due to radiation, and therefore no difference in the economy of steam supplied to the heating apparatus. There is an advantage, however, in carrying a higher boiler pressure and supplying the heating apparatus at reduced pressure to better meet variations in the demands of the heating apparatus and, where necessary, for operation of a pump for boiler feeding or for handling the returns; but where these are unnecessary and there is no demand for high-pressure steam for other operations, it is safer and, on that account, better to operate a heating boiler at no higher pressure than necessary to supply the heating apparatus.

QUESTION [Relative Effect of Low- and High-Pressure Radiators]—*What is the relative effectiveness of dry room radiators supplied with exhaust steam at a pressure of 2 lb. gage and when supplied with live steam at 100 lb. gage?*

B. C.

Answer—When there is complete removal of air from the radiators and perfect circulation of steam, the heat radiated for a square foot of surface may be considered as directly in proportion to the difference of temperature of the steam within the radiator and the room temperature in the immediate vicinity of the radiator. Assuming, for instance, that

the room temperature maintained near the radiator is 90 deg. F., as the temperature of steam at 2 lb. gage is 218.5 deg. F., and the temperature of steam at 100 lb. gage is 338 deg. F., then the relative effectiveness per square foot of radiation would be as (218.5—90) for the exhaust steam, to (338—90) for the steam at 100 lb. gage, or as 100 to 193.

QUESTION [Poor Circulation in Radiator of Vacuum System]—*In a vacuum heating system that supplies heat for two houses, there is heating up of only about one-third of the number of sections of a radiator in the building farther removed from the boiler. What may be the cause and remedy?* R. A.

Answer—The cold sections probably remain airbound due to insufficient vacuum. If it is not possible to obtain a better vacuum at the return while the steam supply is open, the air might be dislodged, sufficiently for starting operation of all sections, by exhausting the radiator with the best vacuum obtainable for ten or fifteen minutes prior to admission of steam.

QUESTION [Measuring Vacuum Without Steam and Vacuum Gage]—*How can inches of vacuum be measured without the use of a vacuum gage or a mercury column?* J. K.

Answer—A steam-engine indicator connected to an engine cylinder or condenser will give the number of pounds pressure per square inch below the pressure of the atmosphere, and that number of pounds of pressure, divided by 0.491 will give the number of inches of vacuum shown by the indicator; or for practical purposes multiply the number of pounds pressure per square inch below atmosphere by 2.

Or hermetically connect the supposed vacuum or partial vacuum with the upper end of a vertical glass tube whose lower end is submerged below the surface of water that is exposed to the pressure of the atmosphere. Then as mercury is about 13.58 times the weight of water, the number of inches of water forced up in the tube above the water surface exposed to atmospheric pressure, divided by 13.58 will be the approximate number of "inches of vacuum" in the condenser or other vessel that is connected with the upper end of the glass tube.

QUESTION [Inches of Vacuum Not Determinable from Temperature]—*Cannot the number of inches of vacuum, or absolute pressure, of a condensing system be known from the pressure corresponding to the temperature of dry saturated steam as shown by the steam tables?* J. H. C.

Answer—The temperature is not an index of the pressure because of the presence of an unknown quantity of "air." The exhaust of an engine is a mixture of water, vapor and gases, the gases being a mixture of those originally dissolved in the boiler-feed water and atmospheric air which leaks into those parts of a condensing system in which a partial vacuum is

maintained. Hence, the absolute pressure within the condenser, as indicated by a vacuum gage, is that of water vapor at the pressure corresponding to the temperature, plus the pressure due to any gases present. As there always is more or less of the gases present, an actual vacuum-gage reading always shows higher absolute pressure than the pressure of saturated steam corresponding to the temperature as given in tables of properties of steam.

QUESTION [Blowdown of Safety Valve]—*What is meant by adjustment of the blowdown of a safety valve?*

W. G. M.

Answer—"Blowdown" is the less number of pounds pressure per square inch at which a safety valve closes than the pressure at which the valve is set to pop open. In most forms of spring pop safety valves, the amount of blowdown may be varied by adjusting the position of a ring, called the blowdown ring, which surrounds the valve seat and deflects the escaping steam in such a manner as to assist in holding the valve open until the boiler pressure has been reduced by the amount of blowdown.

QUESTION [Allowable Blowdown of Boiler Safety Valves]—*What should be the amount of blowdown of a boiler safety valve?* G. N. H.

Answer—The amount of blowdown admissible depends on the boiler pressure that is carried. The A. S. M. E. boiler code stipulates that safety valves on boilers carrying an allowed pressure of less than 100 lb. per sq. in. gage must be adjusted to close after blowing down not more than 4 lb.; when carrying pressures 100 to 200 lb., the blowdown must be not over 6 lb.; and when carrying over 200 lb., the blowdown must not be more than 8 pounds.

QUESTION [Too Much Blowdown of Safety Valve]—*If a pop safety valve that was set at 90 lb. is adjusted to blow at 70 lb. and the valve does not shut off until the steam pressures of the boiler drops to 60 lb., what should be done?* H. A.

Answer—Safety valves of boilers carrying an allowed pressure of less than 100 lb. per sq. in. gage should close after blowing down not more than 4 lb. If the safety valve, when set to blow at 70 lb., does not close until the steam pressure falls to 60 lb., thereby allowing 10 lb. blowdown, the blowdown should be adjusted so the valve will close at not less than $70 - 4 = 66$-lb. pressure.

QUESTION [Testing Safety-Valve Capacity]—*What test can be made for determining whether a safety valve is large enough for a boiler?* D. M.

Answer—To be of sufficient capacity, a safety valve must be capable of relieving the boiler of steam at such a rate that the boiler pressure cannot be raised above the safe working

pressure when the firing is forced as much as possible, and there is no steam escaping except through the safety valve. Such a capacity test should be made by gradually increasing the rate of firing, but should not be made without having means available that are known to be reliable for immediately checking the fire and also for relieving the boiler of steam, in case there is any increase of pressure above the safe working pressure for the boiler.

QUESTION [**Size of Safety Valve for Compressed-Air System**]—*What is the largest safety valve permissible on an air compressor?* L. W. R.

Answer—For safety there would be no limitation of the maximum size. There is, however, a limitation of minimum capacity. The Massachusetts Air-Tank Regulations, which were carefully prepared, specify (a) that the maximum commercial rating sea-level pressure and 60 deg. F. of any air compressor shall be the piston displacement in cubic feet per minute at the maximum speed as given in the catalog of the manufacturers, and (b) the safety valve connected to an air-compressor system shall have, when adjusted to give the maximum discharge area for satisfactory operation, a capacity capable of discharging a quantity of air at least equal to 25 per cent excess of the maximum rating of the air compressors operating on said system, without the air pressure rising over 5 per cent of the allowable pressure.

QUESTION [**Larger Safety Valve for Relieving Lower Boiler Pressure**]—*Can the same safety valve be used if the safe working pressure of a boiler is changed from* 100 *to* 25 *lb. per sq. in.?* R H.

Answer—If the safety valve is no larger than was necessary for carrying 100 lb. pressure, then with all other conditions the same it would be too small for 25 lb. pressure. Steam at the higher pressure is denser, contains more heat per pound and would be discharged at higher velocity through a given valve aperture, and for equivalent relief of heat a larger safety valve is needed with the lower pressure.

QUESTION [**Weight Required at End of Safety Valve Lever**]—*If the area of a lever safety valve is* 1 *sq. in., distance from valve to fulcrum* 2 *in., weight of lever* 2 *lb., distance of its center of gravity to the fulcrum* 5 *in., weight of the valve and stem* ½ *lb., and total length of lever* 15 *in., what weight must be placed on the end of the lever so the valve will blow off for a pressure of* 100 *lb. per sq. in.?* A. B. T.

Answer—The lever alone will exert a pressure on the valve of $(2 \times 5) \div 2 = 5$ lb., and as the weight of the valve and stem is 0.5 lb. the lever and dead weight of the valve will balance a pressure of 5.5 lb., leaving $100 - 5.55 = 94.5$ lb. of pressure to be exerted on the valve by the leverage of the weight placed at the end of the lever. As 1 pound at

the end of the lever would exert a pressure on the valve of $1 \times 15 \div 2 = 7.5$ lb., the additional 94.5 lb. pressure required would be balanced by a weight of $94.5 \div 7.5 = 12.6$ lb. placed on the end of the lever.

QUESTION [**Chatter and Blow-down of Safety Valve**]—*Why does a safety valve continue to blow off after the boiler pressure has dropped below the pressure for which the valve is set? What would cause a safety valve to chatter or rumble when blowing off?* C. T. C.

Answer—Blowing off continues after the boiler pressure is reduced below the "pop" pressure because the discharge passages guide the escaping steam in such a manner that it helps the boiler pressure to hold the valve away from its seat. The greater this assistance while the valve is open the lower the boiler pressure will be reduced below the "pop" pressure before the valve will be closed. The reduction in boiler pressure below the "pop" pressure is called the "blow-down" and most direct spring loaded safety valves are provided with an adjustable "blow-down ring" or other means for varying the amount of blow down. When the valve is raised from its seat the pressure exerted by the spring is greater than when the valve is closed. Hence when the valve opens, unless the escaping stem exerts enough force to overcome the greater spring resistance, for continuance of a little more than pop pressure in the boiler there are rapid alternations of partial openings and closings, due to the ricochetting of the spring and small sudden changes of boiler pressure that give rise to chattering of the valve with intermittances of discharge, causing vibrations of the atmosphere that produce a chattering or rumbling sound.

Section IV
Fuels and Furnaces

QUESTION [General Formula for Excess Air]—*How is the percentage of excess air determined from the percentage of CO_2 in the flue gases?* R. L. S.

Answer—Excess air is admitted in excess of the amount theoretically required to furnish enough oxygen for complete combustion of the fuel, according to its chemical composition, and no general formula can be constructed for determining the per cent of excess air in the flue gases without reference also to the CO and O_2 present. But assuming, as in the case of anthracite coals, that the air supply is principally required for combustion of carbon, an approximation of percentage of excess air based on the percentage of CO_2 can be found by subtracting the observed percentage of CO_2 from 20.7, dividing the remainder by the observed percentage and multiplying by 100. Suppose, for instance, that the CO_2 were 8 per cent, then the approximate percentage of excess air passing through the fire would be $\frac{20.7 - 8}{8} \times 100$, or $\left(\frac{20.7}{8} - 1\right) \times 100 =$ 158.7 per cent.

A more accurate method is to determine the percentage of excess air from the amount of CO_2 and oxygen in the flue gases. A simple alignment chart for the purpose is given on page 568 of October 11, 1921, issue of *Power*.

QUESTION [Necessity for Excess Air]—*Would it not be better to dispense with all excess air admitted to a boiler furnace?* J. W. S.

Answer—Excess air is the amount of air admitted in excess of the amount of air theoretically required for complete combustion. But to insure complete combustion and give each atom of carbon and other combustible elements opportunity to combine with the necessary amount of oxygen to effect complete combustion it is necessary for the air supply to be in excess of the theoretical requirement. When only enough excess air has thus been supplied to obtain a high percentage of CO_2 in the flue gases the heat obtained from the combustion exceeds the heat that would be obtained from imperfect combustion with no excess air by an amount considerably greater than the loss incurred from such excess air.

QUESTION [Heat Loss from Moisture in Coal]—*What effect has moisture in coal on its heat value and how is the loss determined?* W. R. N.

Answer—Water in coal must be evaporated and superheated and escapes at the flue temperature at atmospheric pressure. Heat is thus rejected to the chimney and therefore any moisture in coal, in addition to being a dead weight, results in waste of heat. The heat can be determined by the formula,

$$\textit{Loss in B.t.u. per pound of dry coal} = M\,[212 - t + 970.4 + 0.47\,(T - 212)]$$

in which

M = Pounds of moisture per lb. of dry coal;
t = Room temperature;
T = Temperature of the flue gases;
970.4 = Latent heat of evaporation f. and a. 212 deg. F.;
0.47 = Mean specific heat of superheated steam.

For example, assume a coal of 13,000 B.t.u. (dry basis) 8 per cent moisture, 60 deg. F. temperature of coal as fired and 550 deg. F. stack temperature. Then by substituting in the formula, the heat loss is found to be $0.08\,[212 - 60 + 970.4 + 0.47\,(550 - 212)] = 102.5$ B.t.u. per lb. of coal, or, $102.5 \times 100 \div 13{,}000$ = about 0.8 of one per cent of the heat in the dry coal, from which it may be noted that with small percentages of moisture the loss amounts to approximately 1 per cent for each 10 per cent of moisture.

QUESTION [Weight of Discharged Chimney Gases]—*If a ton of carbon is burned in a furnace using 20 lb. of air per pound of fuel, how many pounds of carbon dioxide, free oxygen and nitrogen would be discarded up the chimney, assuming that the rare gases are classed with the nitrogen?* R. L. T.

Answer—A ton of carbon would be 2,000 lb. To burn a pound of carbon to CO_2 requires 2.66 lb. of oxygen. Hence in burning 1 lb. of carbon to CO_2 we send up the chimney 1 lb. of carbon + 2.66 lb. of oxygen = 3.66 lb. of CO_2, and for 2,000 lb. of carbon there would be $2{,}000 \times 3.66 = 7{,}320$ lb. of CO_2 discharged up the chimney.

Air contains 0.23 parts by weight of oxygen and 20 lb. of air would contain $20 \times 0.23 = 4.6$ lb. of oxygen, so that the total oxygen supplied to burn 2,000 lb. of carbon would be $2{,}000 \times 4.6 = 9{,}200$ lb. of oxygen. But as combustion of the carbon required only $2{,}000 \times 2.66 = 5{,}320$ lb. of oxygen there would be $9{,}200 - 5{,}320 = 3{,}880$ lb. of free oxygen discharged up the chimney.

As air contains 0.77 parts nitrogen by weight, there would be $2{,}000 \times 20 \times 0.77 = 30{,}800$ lb. of nitrogen received and discharged up the chimney.

QUESTION [**Checkerwork for oil burners**]—*Is it visable to use a brick checkerwork in an oil-burning furnace?*

Answer—The checkerwork is useful in breaking up the oil spray where atomizing is not perfected at the burner tip. A well-designed burner should cause an oil and air mixture without the checkerwork although it may be useful in re-igniting the fuel after a momentary blow out.

QUESTION [**Formula for Estimating Heat Value of Coal**]—*How is the heat value of coal estimated from the chemical analysis of the coal?* D. A. B.

Answer—The heat value of coal may be calculated from its ultimate analysis with a probable error not exceeding 2 per cent, by Dulong's formula:

$$\textit{Heat value in B.t.u. per pound} = 146\ C + 620\ \left(H \frac{0}{8}\right) +$$

40 S, in which C, H, O, and S are respectively the percentages of carbon, hydrogen, oxygen and sulphur shown by the analysis.

QUESTION [**Heat Energy Realized per Pound of Coal**] —*What per cent of the heat energy in the fuel is realized by an engine in a plant where the coal used contains* 12,000 *B.t.u. per pound, the evaporative economy of the boilers is* 7 *lb. of water per pound of dry coal, the economy of the engine is* 32 *lb. of water per i.hp., and the mechanical efficiency of the engine* 88 *per cent?* R. H. D.

Answer—With evaporation of 7 lb. of water per pound of coal, the fuel consumption would be $32 \div 7 = 4.57$ lb. or coal per i.hp.-hr. and for 88 per cent mechanical efficiency, the energy realized would be $\frac{33,000 \times 60}{4.57} \times 0.80 = 346,608$ ft. lb. or $346,608 \div 778 = 445.5$ B.t.u. per pound of the coal; and for coal containing 12,000 B.t.u. per pound, the per cent of heat energy realized would be $445.5 \times 100 \div 12,000 =$ 3.7 per cent.

QUESTION [**Conversion of B.t.u. Into Horsepower**]—*How many horsepower will be developed by an expenditure of* 210 *B.t.u. per minute?* S. C.

Answer—The mean value of trustworthy investigations for determination of the mechanical equivalent of heat, founded on experiments in which mechanical work was transformed directly into heat and electric energy was changed into heat, when reduced to mean B.t.u., gives the result,

$$1\ \textit{mean B.t.u.} = 777.54\ \textit{standard ft.-lb.}$$

For ordinary calculations the value 777.5 ft.-lb. is taken.

As a horsepower is development of energy at the rate or 33,000 ft.-lb. per min., an expenditure of 210 B.t.u. per min. would develop $(777.5 \times 210) \div 33,000 = 4.95$ *hp.*

QUESTION [Reduction of B.t.u. per Pound to Calories per Long Ton]—*What would be the number of calories per ton of 2,240 lb. of coal containing 11,500 B.t.u. per pound?*
B. H. C.

Answer—The gram-calorie, or small claorie, is the heat required to raise the temperature of 1 gram of pure water from 15 to 16 deg. C. The kilogram-calorie, or large calorie, which is equal to 1,000 gram-calories, is the one generally employed by engineers and is equal to 3.968 B.t.u.; that is, 1 B.t.u. = 0.252 kilogram-calorie. Hence a long ton, or 2,240 lb. of coal containing 11,500 B.t.u. per pound would contain $2,240 \times 11,500 \times 0.252 = 6,491,520$ kilogram-calories.

QUESTION [Proximate and Ultimate Analysis of Coal]—*What is the difference between proximate analysis and ultimate analysis of coal?*
W. N. S.

Answer—The proximate analysis assumes coal to contain four different and separate ingredients called fixed carbon; volatile hydrocarbon or volatile matter or simply volatile; moisture; and ash. The percent of moisture is determined by maintaining a small quantity of finely ground coal at a temperature of about 200 deg F. The material lost in this manner is assumed to be moisture and is reported as such. Volatile matter is determined by heating a sample from which the moisture has been driven or a fresh sample, for which purpose the coal is held at a red to white heat, with exclusion of air until there is no further loss of weight. Fixed carbon is found by combustion of a sample from which the moisture and volatile have been driven off; and ash is the designation of the incombustible material left after determining the fixed carbon. The sum of the volatile, hydrocarbons, and fixed carbon is called the combustible. When sulphur is to be reported it must be determined by a separate analysis. The ultimate analysis is a thorough chemical analysis for determination of the percentages of carbon, hydrogen, oxygen, nitrogen, and sulphur, and the percentage of ash in dry coal.

QUESTION [Loss from Combustible Remaining in Ashes]—*What is the loss from combustible remaining in ashes when the coal as fired contains 12,890 B.t.u. per lb. and 12.6 per cent. ash, if the refuse contains 40 per cent. combustible and the coal costs—6.75 per ton?*
J. A. P.

Answer—Each 100 lb. of coal containing 12.6 per cent. ash would consist of $(1-0.126) \times 100 = 87.4$ lb. of combustible and 12.6 of ash. If the refuse from firing consisted of 40 per cent. combustible and $100 - 40 = 60$ per cent. ash, then the refuse from firing 100 lb. of coal would contain $12.6 \div 60 \times 40 = 8.4$ lb. of combustible, and the loss of combustible contained in the coal would be $(8.4 \times 100) \div 87.4 = 9.6$ per cent. Hence, if the coal contained 12,890 B.t.u. per lb. of combustible, there would be a loss of 9.6

per cent. of 12,890 = 1237 B.t.u. per lb. of coal; and with coal costing—6.75 per ton there would be a loss of 9.6 per cent. of—6.75 = \$0.65 per ton of coal used.

Expressed algebraically, if a = percentage of ash in the coal and c = percentage of combustible in the refuse, then $100 - c$ = percentage of ash in the refuse and $\left(\frac{a}{100-c}\right) \times c$ $100 \div (100 - a)$ = percentage of loss. If as in the example, $a = 12.6$ and $c = 40$, by substitution the formula becomes $\left(\frac{12.6}{100-40} \times 40\right)100 \div (100-12.6) = 9.6$ per cent. loss.

QUESTION [Carbon Mineral or Mined Coal]—*We have an inquiry from a Central American customer for a boiler that is to be provided with grate suitable for burning "mineral coal." What particular kind of fuel is referred to?* S. H.

Answer—In Spanish-American countries, extensive use is made of charcoal (carbon de leña) and your correspondent intends to confine consideration to mined coal, for which the Spanish technical designation is "carbon mineral" and the common designation is "carbón de piedra," or stone coal. The inquiry undoubtedly refers to use of a low grade of bituminous coal.

QUESTION [Relative Value of Coal Deprived of Volatiles]—*Having coal containing* 12,000 *B.t.u. per lb. and* 30 *per cent volatile, what would be the loss if burned to a point where the volatile is consumed and then extinguished for use as fuel elsewhere; and what would the remaining coke be worth per ton if the original coal was worth* \$6 *per ton?* H. G. E.

Answer—Coking of bituminous coal in modern beehive ovens yields about 60 per cent of the original weight as marketable coke with expulsion of about 30 per cent of the volatile matter from the coal. All the volatile could not be driven off without the loss of such a high percentage of total combustibles as to render the coking process very extravagant. The percentage of weight retained would depend on the coking process employed, and the value of the residue for fuel would depend on its analysis.

QUESTION [Burning Mixed Sizes of Coal]—*For hand-firing boilers, is it better to stoke mixed sizes of coal or fire with one size at a time?* R. N. G.

Answer—The best results are obtained when there is greatest uniformity in size of the coal. When large and small sizes are mixed together in equal proportions there is more obstruction to the draft than from use of only a smaller size, as the fire becomes dirty before the larger sizes have been consumed and in cleaning the fire there is greater waste of unburned coal. Greater economy is to be obtained by suitably firing the different sizes separately, and if that is not

practicable for obtaining the necessary boiler capacity the different sizes should be stoked in alternate layers with changes of size and thickness of the layers so timed as to obtain complete combustion.

QUESTION [**Heat of Combustion of Various Fuels**]—*What quantity of heat is developed by combustion of fuel oil, coal, natural gas, carbon, hydrogen and acetylene gas?* N. C. D.

Answer—The heat of combustion of fuel oil coal and natural gas depends on their analyses. The heat of combustion of fuel oils varies from 18,000 to 20,000 B.t.u. per lb.; commercial coal from 10,000 to 14,000 B.t.u. per lb.; natural gas about 1000 B.t.u. per cu. ft. The heat of perfect combustion of pure carbon is about 14,500 B.t.u. per lb.; perfect combustion of hydrogen in air 50,000 to 60,000 B.t.u. per lb. and acetylene in perfect combustion, with oxygen, 1685 B.t.u. per cubic foot.

QUESTION [**Coal Required for Heating Air**]—*A fan delivers 36,000 cu. ft. of air per minute through steam-heating coils. What amount of coal of 10,000 B.t.u. per lb. will be required to raise the temperature of the air from 10 deg. F. below zero to 70 deg. above zero?* C. C. P.

Answer—The specific heat of dry air at constant pressure is about 0.2376 and at − 10 deg. F. the weight per cubic foot would be about 0.0882 lb. As the elevation of temperature would be 80 deg., the heat required would be $36,000 \times 60 \times 0.0882 \times 0.2376 \times 80 = 3,621,252$ B.t.u. per hour. Allowing a boiler efficiency of 60 per cent. to be obtained with coal of 10,000 B.t.u. per lb., would require, $3,621,252 \div (10,000 \times 0.6) = 604$ lb. of coal per hour.

QUESTION [**Air Required for Burning Coal**]—*How much air must be supplied to burn a pound of coal?* H. L. B.

Answer—The air required for the combustion of a pound of coal is given approximately by the formula,

$$\textit{Weight of air in pounds} = 12\,C + 35\,(H - \frac{O}{8})$$

in which C, H and O represent the parts of a pound of carbon, hydrogen and oxygen respectively in a pound of the coal. For perfect combustion most fuels combine with about 12 lb. of air per pound of the fuel. But in burning the fuel, to insure that each atom of carbon will meet with an abundance of oxygen for complete combustion, more air must be admitted to the furnace than is necessary for the combination. The weaker the draft the more excess air will be required for this purpose. With natural draft it is usual to provide for a supply of about 100 per cent excess air, and with forced draft about 50 per cent. As a pound of air at 62 deg. F. has a volume of 13.14 cu. ft., with natural draft the provision for air supply should be about $12 \times 2 \times 13.14 = 315$ cu. ft., and

with forced draft about $12 \times 1.5 \times 13.14 = 236$ cu. ft. per pound of coal that is to be burned. Then, for best economy, the quantity of air actually supplied should be no more than required under the conditions for obtaining the highest percentage of CO_2 in the flue gases.

QUESTION [Coal Chargeable to Use of Closed Heater]—*How much coal should be charged per* 100 *cu. ft. of water, heated from* 60 *to* 200 *deg. F. in a closed exhaust-steam heater?*

W. S. S.

Answer—The amount of coal chargeable will vary in different cases. The service should be charged with the cost of any increase of average back pressure on the engines; or on the other hand, if the heater acts like a condenser of the exhaust and reduces the back pressure, then the service should be credited with the improvement of economy resulting from reduction of the average engine back pressure. The increase or decrease of average back pressure on the engine, due to use of the heater, should be determined from indicator diagrams of the engine taken with the average load before and after the heater is installed. Then multiplying the pounds of coal originally required per pound m.e.p. per hour by the variation in back pressure will give the quantity of coal to be debited or credited to use of the heater.

QUESTION [Percentage Improvement in Coal Economy per Kilowatt-Hour]—*Our fuel consumption for October was* 1,960,010 *lb. of coal costing* \$7.44 *per ton of* 2,240 *lb. for an electrical output of* 524,270 *kw.-hr. For December the fuel consumption was* 1,746,445 *lb. of coal costing* \$7.44 *per ton of* 2,240 *lb., and* 690,260 *lb. of screenings costing* \$1.50 *per ton of* 2,000 *lb. for an electrical output of* 574,040 *kw.-hr. What was the percentage of improvement in coal economy?* G. A. E.

Answer—For October the fuel cost was

$$\frac{\$1{,}960{,}010 \times -7.44}{2{,}240} \div 524{,}270 = -0.01242 \textit{ per kw.-hr.}$$

For December the fuel cost was

$$\left[\frac{1{,}746{,}445 \times \$7.44}{2{,}240} + \frac{690{,}260 \times \$1.50}{2{,}000}\right] \div 574{,}040 = \$0.01101 \textit{ per kw.-hr.}$$

Hence for December the improvement in coal economy over that for October was

$$\frac{(\$0.01242 - \$0.01101) \times 100}{0.01242} = 11.35 \textit{ per cent.}$$

QUESTION [Banking Fire with Coarse Coal]—*We have been accustomed to use run-of-mine bituminous coal, employing the smallest sizes for banking fires. The last coal shipment received was nearly uniform stove size, and we are troubled to hold a banked fire. How should the banking be done?*

Answer—Prepare a heavier bank of clean live coals sloped against the bridge-wall and covered with a layer of fresh coal. To hold the fire, blanket the bank with a thick enough layer of fine ashes to prevent a draft through the coals. For restarting the fire, drag off the ashes and only enough of the top layers of coal to leave a bank of clean kindling coal.

QUESTION [Holes for Inspecting Thickness of Corrugated Furnaces]—*What means should be provided for inspecting the thickness of a corrugated furnace?* F. S. J.

Answer—For ascertaining the thickness of the material of a corrugated or ribbed furnace by actual measurement, the furnace should be drilled for a $\frac{1}{4}$-in, pipe tap and fitted with a removable screw plug. For Brown and Purves furnaces boiler codes usually provide that the holes should be in the center of the second flat, and for Morrison and other similar types, in the center of the top corrugation, at least as far in as the first corrugation.

QUESTION [Less Grate Area with Stokers than with Hand Firing]—*Why are the grate areas of stokers usually made less than the areas of hand-fired grates for the same furnaces?* W. N. J.

Answer—The motion of the stoker maintains a cleaner fire by constantly disturbing the film of ash formed by the burning coal. The rate of combustion attainable per square foot of grate is thus increased to an extent that more than compensates for the smaller grate area.

QUESTION [Coefficient of Expansion of Firebrick]—*What is the coefficient of expansion of firebrick?* F. E. C.

Answer—Experiments have shown that the coefficient of expansion of firebrick decreases with temperature rise, but an average coefficient of linear expansion within the range of temperatures ordinarily attained in boiler furnaces would be about 0.0,000,027 per degree F.

QUESTION [Dull Black Finish for Brasswork]—*What treatment is given brasswork to impart a dull black finish?*
A. W. B.

Answer—Make a concentrated solution of copper in nitric acid and add an equal quantity of water. The surfaces to be blackened should be bright and washed clean of grease. If the article is large, swab the surfaces with the diluted copper solution; if small, immerse it in the solution. After draining or shaking off the solution, heat the article until the copper salt is converted to a black oxide. The heating may be done over a clean coal fire, but better results will be obtained by heating the articles in a closed muffle furnace, so as to obatin uniformity of heating and coloring. For some kinds of work more uniform results may be obtained

by using a weaker solution, as one part of the concentrated solution to two parts of water.

QUESTION [Coefficient of Expansion of Mercury]—*What is the coefficient of expansion of mercury per degree Fahrenheit?* J. V. H.

Answer—For practical purposes the cubical expansion of mercury may be taken as 0.0001 per degree Fahrenheit. The coefficient increases with the temperature. According to determinations made by Regnault, the mean coefficient between 32 deg. F. and 212 deg. F. is 0.00010086; between 212 and 392 deg F. it is 0.00010338; and between 392 deg. F. and 572 deg. F. it is 0.00010646.

QUESTION [Power Required To Drive an Emery Wheel]—*Will you publish the formula for figuring the power required to drive an emery wheel?* G. L.

Answer—There is no method of figuring the power required to drive an emery wheel. However, the power required will depend upon the size of the wheel and the kind of work it has to do. Experience has shown that, ordinarily, a 6-in. wheel requires from 0.5 to 1 hp.; a 10-in. wheel, 2 hp.; a 12-in. wheel, 3 hp.; an 18-in. wheel, 5 to 7.5 hp.; and a 24-in. wheel, 7.5 to 10 horsepower.

QUESTION [Weight of Compressed Air]—*What is the weight of 24 cu.ft. of dry air at a temperature of 125 deg. F. under a pressure of 26 lb. per sq. in. gage?* E. C.

Answer—At atmosphereic pressure, or 14.696 lb. per sq. in. absolute, a pound of dry air at 32 deg. F., or an absolute temperature of $460 + 32 = 492$ deg., has a volume of 12.387 cu. ft.; and as the density is inversely as the absolute temperature and directly as the absolute pressure, the weight of 12 cu. ft. of air at the temperature of 125 deg F., or $460 + 125 = 585$ deg. F., and pressure of 26 lb per sq. in. gage, or $26 + 14.696 = 40.696$ lb. per sq. in. absolute, would be

$$\frac{492 \times 40.696 \times 24}{585 \times 14.696 \times 12.387} = 74 \text{ lb.}$$

QUESTION [Horsepower Required to Raise Water]—*What number of horsepower would be required ro raise 2,500 gal. of water to a height of 60 ft.?* W. S. R.

Answer—Horsepower is a rate of doing work and the number of horsepower required to be developed while raising 2,500 gal. of water to an elevation of 60 ft. at a uniform rate would depend on the time employed. A gallon of water weighs 8.33 lb., and 2,500 gal. would weight $2{,}500 \times 8.33 = 20{,}833$ lb. If the pumpage of 2,500 gal. is performed in one hour, then there would be $20{,}833 \div 60 = 347.2$ lb. of water pumped per minute; and if elevated 60 ft. the net work would be $347.2 \times 60 = 20{,}833$ ft.-lb. per minute. As one horsepower is development of work at the rate of 33,000 ft.-lb. per minute

the net power developed would be at the rate of 20,833 ÷ 33,000 = 0.6. hp. Neglecting pipe friction, the power required to be developed on the steam end of a pump to develop an actual or net horsepower, would depend on the mechanical efficiency of the pump. For pumps of the size required the mechanical efficiency would be about 60 per cent, and the power required to be developed on the steam end for development of 0.6 hp. in the water end would be about 0.6 ÷ 0.6 = 1 i.hp.

QUESTION [**Compensating Variation from Scale of Planimeter**]—*How is a planimeter used for measuring the mean effective pressure shown by an indicator diagram made with a 16-lb. spring where the planimeter scale is intended to show m.e.p. for a 30-lb. spring?* C. L. J.

Answer—Operate the planimeter as though a 30lb. spring had been used and take 16/30 of the result.

QUESTION [**Wear and Breakage of Chain Drive of Ventilation Fan**]—*Trouble is experienced with wear and frequent breakage with a chain drive used for transmission of power from a motor to a 16-ft. diameter ventilating fan of a mine. The pitch of the chain is 2 in., the sprocket wheel on the fanshaft is about 5 ft. 5 in. pitch diameter with 102 teeth, the pinion on the motor has 19 teeth and the chain has 124 links. What may be the cause of the breakdowns?* V. J. S.

Answer—It is likely that the distance between centers and form of teeth are not adapted to the pitch of the links and relative sizes of the wheels, causing delivery of the slack side of the chain out of mesh with the sprocket on the fanshaft. The data given indicate that the center-to-center distance of the sprockets is about 50 inches. To insure good operation, the distance should not be less than about 40 times the pitch or 80 inches, and it is better to have the slack side of the chain at the bottom.

QUESTION [**Left-Hand Adjusting Screw of Monkey Wrench**]—*Why are monkey wrenches usually provided with left-hand adjusting screws?* T. W.

Answer—A left-hand adjusting screw affords greater convenience in closing up the jaws of the wrench by operation of the milled collar with the thumb, while the wrench is held in the grasp of the fingers of the right hand.

QUESTION [**Inch of Water and Pressure per Square Inch**]—*What is the value of an inch of water pressure expressed in pounds or ounces per square inch?* W. D. J.

Answer—The pressure exerted per inch of depth of an incompressible liquid is the same pressure per square inch as the weight of a cubic inch of the liquid. A cubic foot of water at standard temperature (62 deg F.) weighs 62.355 lb., and assuming water to be incompressible, an inch of

water pressure is equal to $62.355 \div 1728 = 0.03609$ lb. per sq. in. or $0.03609 \times 16 = 0.5774$ oz. per square inch.

QUESTION [Transmissive Capacity of Steel Shafting]—*What is the rule for estimating the approximate horsepower-transmitting capacity of the ordinary sizes of steel shafting?*

S. M.

Answer—The approximate number of horsepower capable of being transmitted with safety by ordinary sizes of turned steel lineshafting, when well supported, with pulleys near to the bearings and with the hangers so spaced that the deflection will not exceed 0.01 in. per foot of span, is equal to the cube of the diameter of the shaft in inches for 100 r.p.m. and directly in proportion for other speeds. The safe load for head shafts may be taken as 75 per cent. and for bare transmission shafts as 175 per cent. as much as for ordinary line shafting. Cold-rolled shafting may be taken as one-third stronger than turned steel shafting.

QUESTION [Improving Soiled Tracings for Blue-printing]—*How can tracings, spoiled and crumpled from use, be improved for obatining blueprints?* H. T. T.

Answer—Most of the grease and dirt can be removed by lightly rubbing off the tracing with a tuft of clean absorbent cotton moistoned with gasoline. After being cleaned and dried the tracing can be flattened out by ironing the back with a sadiron, which should not be so hot as to impair the transparency of the material.

QUESTION [Data for Ordering Pinion]—*If the pinion to an elevator should be broken beyond repair, what data should be sent to the factory to secure a wheel to replace the broken one?*

W. H.

Answer—When ordering of the maker, state the purpose of the wheel, and give the pattern number if it bears one. The order should state whether a cut or cast gear is to be supplied. Give an accurate measurement of the working distance between the center of the pinion and center of gear wheel with which the pinion meshes, and the number of teeth and outside diameter of each, and furnish a drawing or templet the shape of a segment of the gear wheel that will include eight or ten teeth, showing their shape marked off from the wheel on a sheet of tin plate or cardboard, and state the desired width of face of the teeth for the pinion. Also state the length of hub, how much each end of the hub is to be offset under or beyond the teeth; whether ends of hub are to be finished, diameter of bore, size of setscrew, size of keyway in shaft, whether new key is wanted and from which end of the hub key is to be driven.

QUESTION [Trouble with Hand Rope of Elevator]—*We have considerable trouble from the hand rope on a freight*

elevator becoming too tight or too slack for use within an hour or two after the length has been adjusted, and find the rope has turned the swivel at the turnbuckle. What is the cause and remedy? E. P.

Answer—This trouble is common with wire hand ropes for operating elevators, especially where long ropes are required for elevators passing through a number of stories. Tightening and slackening of the rope is due to expansion and contraction of the wire from changes of temperature and swelling and shrinkage of the hemp core due to variable moisture of the atmosphere. For most situations the use of tightener pulleys for automatically preserving the tension would be impracticable and it is necessary to depend on hand adjustment of a turnbuckle. The bother occasioned by a slackened hand rope coming off the sheaves may be avoided by providing guards each side of the sheaves.

QUESTION [Number of Cubic Yards in Block of Concrete]—*How many cubic yards of concrete are contained in an engine foundation* 4 *ft. x* 12 *ft. at the top*, 6 *ft. x* 14 *ft. at the base and* 5 *ft.* 6 *in. deep, with sides and ends battered?*

H. W. G.

Answer—Taking all dimensions in feet, the content in cubic feet is given by the prismoidal formula:

Number of cu. ft. = (area of top + area of base + 4 times sectional area at mid depth) ÷ 6 × total depth.

The area of the top would be $4 \times 12 = 48$ sq. ft., and that of the base would be $6 \times 14 = 84$ sq. ft. The section at mid-depth would be $\frac{4+6}{2} \times \frac{12+14}{2}$, or 5 ft. × 13 ft., and the area of the mid-depth section would be 65 sq. ft. Substituting these values and 5.5 ft. for total depth, the prismoidal formula gives,

$$\text{Number of cu. ft.} = \frac{48+84+(4\times65)}{6} \times 5.5 = 359.33 \text{ cu. ft., or}$$

$$359.33 \div 27 = 13.31 \text{ cu. yd.}$$

QUESTION [Pull and Power Required to Draw Car Up Incline]—*In a balanced monitor incline system it is required to raise a load of* 7,500 *lb. up an incline* 750 *ft. long, a total elevation of* 150 *ft. The speed of rope carrying the load is* 600 *ft. per min. There will be two cars, one being pulled up the incline loaded, while the other returns empty. Neglecting friction, air resistance and weight of cars, what will be the pull on the rope and what horsepower will be required?* A. F.

Answer—When the car is held at rest or is being drawn up the incline at a uniform speed, the pull on the rope would amount to $^{150}/_{750}$ of 7,500 lb. = 1,500 lb., and while drawn up the incline at the uniform speed of 600 ft. per min., the power required would be $600 \times 1,500 \div 33,000 = 27.27$ hp. For moving the car up the incline from a state of rest, there

would be an additional pull on the rope, depending on the acceleration. If the time allowed for increasing the speed from 0 to 600 ft. per min., or 10 ft. per sec., was 15 sec., with uniform acceleration, then the acceleration would be $10 \div 15 = \frac{2}{3}$ ft. per sec. Forces are to each other as the accelerations produced. The acceleration due to gravity is 32.16 ft. per sec., and therefore the force required to produce the acceleration only would be $7{,}500 \times \frac{2}{3} \div 36.16 = 155.4$ lb. and the whole pull on the rope while raising the speed of the car to 600 ft. per min. would be $1{,}500 + 155.4 = 1{,}655.4$ lb.

QUESTION [Conversion of Inches of Vacuum to Pounds per Square Inch]—*What pressure per square inch would be equivalent to* 26 *in. vacuum?* E. C. H.

Answer—A column of mercury 1 in. high at ordinary temperature exerts a pressure of 0.491 lb. per sq. in., and "inches of vacuum" signifies the number of inches of mercury-column pressure less than the pressure of the atmosphere at the place of observation. When a vacuum gage indicates 26 in. vacuum, the pressure is $26 \times 0.491 = 12.766$ lb. per sq. in. less than the pressure of the atmosphere. The absolute pressure of the atmosphere varies with the elevation of the place. At sea level it usually is assumed to be 14.7 lb. per sq. in., so that at sea level the absolute pressure for 26 in. vacuum would be, $14.7 - 12.766 = 1.934$, or practically 2 lb. per sq. in. absolute.

QUESTION [Regrinding Plug Cocks]—*What is the best method of grinding small brass plug cocks in place? Grinding the plug in its seat with a carpenter's brace has been tried with poor results.* F. L. W.

Answer—When the plug is badly cut, it may have to be skimmed up in a lathe, or if the seat is badly cut, it will need reaming out with a tool that has the correct taper. For grinding, smear the plug with oil and fine emery or, better, use oil and pulverized glass. A better job of grinding is done by oscillating the plug by hand, occasionally turning it to a different position, each time cleaning off the plug and seat and applying a thin even coating of the grinding material. The operation is a tedious one, but is necessary for making a good job of grinding the plug and seat to good conical form. Grinding by turning the plug clear around, as with a brace, is likely to cause scoring of the surfaces.

QUESTON [Material for Reinforcing Concrete]—*Cannot old hoisting cable be used to good advantage for reinforcing a concrete foundation for an engine and direct-connected generator?* W. A. T.

Answer—Reinforcing material for concrete is principally of value in resistance of initial tensile and shearing stresses, for which purpose wire cable would be of little advantage in prevention of cracks or for maintaining the solidity of a

foundation. Cable would be useful mainly for preventing sections of the concrete from falling apart after fracture had occurred and, for that purpose might be useful for incorporation in concrete work where permanent monolithic construction is not imperative.

QUESTION [General Dimensions of Overshot Waterwheel]—*What should be the diameter, general dimensions and speed of an overshot waterwheel with flat-sided buckets, to develop* 5 *brake horsepower with* 3000 *gallons of water per minute?*
R. D. M.

Answer—The weight of water supplied per minute would be $3000 \times 8.33 = 24{,}990$ lb. A well-designed wheel with flat-sided buckets would have an efficiency of about 50 per cent. For development of 5 b.hp. the energy of the water would need to be $(5 \times 33{,}000) \div 0.5 = 330{,}000$ ft.-lb. per min., and the water would need to be supplied with a total head of not less than $330{,}000 \div 24{,}990 = 13.2$ ft. The surface velocity of the wheel should be about 6 ft. per sec. and the velocity of water supplied on the wheel should be double that, or about 12 ft. per sec., which is due to a head of about 2 ft. 3 in. If the wheel is to receive the water exactly at the top, the summit of the buckets should not be more than 2 ft. 3 in. below the top water level in the penstock, and the wheel should not be less than $13.2 - 2.25 = 10.95$, or about 11 ft. diameter at the periphery of the buckets. Allowing for operation with 3 in. lift of the spout gate and discharge upon the wheel of 3000 gal., or $3000 \div 7.48 = 401$ cu.ft. of water per min., at a velocity of 12 ft. per sec., or 720 ft. per min., would require a gate and inside width of spout of $401 \div (720 \times 3/1) = 2.23$ ft., or about 27 in., and about 34 in. length of the waterwheel buckets between the shrouds. For a surface velocity of the periphery of the buckets of 6 ft. per sec., or one-half of the spouted velocity of the water, an 11-ft. wheel should be geared for a speed of $(6 \times 60) \div (11 \times 3.1416)$, or about $10\frac{1}{2}$ revolutions per minute.

QUESTION [Right-Hand and Left-Hand Shoveling]—*Is a right-hand shoveler one who supports the weight with the right hand, or with the left hand?*
G. A.

Answer—A right-hand shoveler supports the weight on the scoop or blade by holding up the tang or the neck of the shovel handle with the left hand. He grasps the handle farther away from the tang, usually at or near the free end of the handle, with the right hand, and the direction of his toss is from his right hand toward his left hand. When the positions of his hands and direction of his toss are the reverse, the method is called left-hand shoveling. Each method is named to accord with the manner that is more natural to a right-handed or to a left-handed person in the exertion of his strength to fill the shovel and perform the toss.

QUESTION [Increasing Length of Bearing May Reduce Friction]—*Does increasing the length of a bearing add to the power required to overcome friction?* W. A.

Answer—For the same materials and finish and same lubrication, up to the point of cutting or abrasion of one surface or the other, the loss of power by friction is directly as the total pressure and independent of the area of the rubbing surfaces. Therefore increasing the length of a bearing does not add to the power required to overcome friction, but on the other hand may reduce the resistance by prevention of cutting or permitting of better lubrication from reduction of the pressure per unit of area of the rubbing surfaces.

QUESTION [Dry Materials for Extinguishing Fire]—*What material could be used in dry powdered form for extinguishing fire?* L. S. E.

Answer—Pulverized bicarbonate of soda (baking soda) or pulverized salt are good materials for extinguishing fire. Dry sand is a good material for smothering small oil fires.

QUESTION [Critical Speed of Shaft]—*What is meant by the critical speed of a shaft?* R. E. B.

Answer—The critical speed is that speed of rotation at which the elastic forces are completely neutralized so that the shaft is incapable of offering any resistance to a deflecting force. This speed is numerically the same as the frequency of vibration that the shaft would have with the same masses mounted on it, if deflected by an external force, while the shaft otherwise is at rest. The axis of rotation rarely passes through the center of gravity, and the "hammering" due to centrifugal deflection is greatest when the r.p.m. are in unison with the number of vibrations.

QUESTION [Submergence for Air Lift]—*What should be the amount of submergence of air pipe for an air lift of a well 275 ft. deep, having water 22 ft. from the surface?* W. G. S.

Answer—Most efficient operation of air lifts usually is obtained with an air-pipe submergence (when the pump is in operation) that will be about 60 per cent of the total air lift, measured from the foot of the air pipe to the level at which the water is discharged, which is equivalent to a submergence $1\frac{1}{2}$ times the height of the point of discharge above the working level of water in the well.

QUESTION [Airbound Water-Supply Line]—*We have a $2\frac{1}{2}$-in. gravity water-supply line about 1,000 ft. long that becomes airbound whenever the line is drawn upon during a few hours to its fullest capacity. How can the trouble be remedied?* A. R. H.

Answer—Liberation of air results from reduction of pressure of the water, especially at high points in the line where excessive drafts produce a siphonage. The remedy would be to

place vent-cocks in the top of the pipe at such points. The vents should be left open a very little all the time. If the slight waste of water is objectionalbe, it can be prevented by connecting the vents to small-sized standpipes extended higher than the level of the water in the supply reservoir.

QUESTION [Semi-Steel]—*What is the material that is known as semi-steel?* R. N. B.

Answer—So-called semi-steel is a special kind of cast iron produced by adding to the molten cast-iron scraps of wrought iron of steel by which the strength and toughness of the cast iron is increased, making it suitable for many purposes where more expensive material would be required.

QUESTION [Alloy for Metallic Packing Rings]—*What is a good composition of soft metals for piston-rod packing rings?* W. B. R.

Answer—For casting metallic packing rings good results have been obtained, using an alloy consisting of lead, $83\frac{1}{3}$ per cent; tin, $8\frac{1}{3}$ per cent; antimony, $8\frac{1}{3}$ per cent. For use with superheated steam tin should be omitted on account of its low melting point with the alloy composed of 80 per cent lead and 20 per cent antimony.

QUESTION [Capacity of Wooden Tank]—*What quantity of water would be contained by a wooden tank having inside measurements of* 18 *ft. diameter at bottom,* 17 *ft. diameter at the top and* 14 *ft. deep?* J. B. A.

Answer—The volume is given by the prismoidal formula; namely *Volume* = 102 *depth* × (*area of top* + *area of the bottom* + 4 *times the cross sectional area at mid-depth*). The area of 17 ft. diameter = 17 × 17 × 0.7854 = 226.98 sq. ft.; the area of 18 ft. diameter = 18 × 18 × 0.7854 = 254.47 sq. ft., and as the diameter at mid-depth would be (18 + 17) ÷ 2 = 17.5 ft., the area at mid-depth would be 17.5 × 17.5 × 0.7854 = 240.53 sq. ft., and by substitution the formula becomes

Volume = $^{14}/_{6}$ [226.98 + 254.47 + (4 × 240.53)] = 3368.33 cu. ft., or 3368.33 × 7.48 = 25,195 gal.

QUESTION [Water Delivered by 6-In. Pipe Line]—*With* 25 *ft. head what quantity of water can be delivered through a* 6-*in. cast-iron pipe* 1000 *ft. long?* T. J. C.

Answer—The highest rate of flow will be obtained when the whole head is absorbed in overcoming friction at the entrance and pipe friction and in producing the velocity with delivery from the discharge end at atmospheric pressure. With 25 ft. head available a clean 6-in. cast-iron pipe line 1000 ft. long, with delivery at atmospheric pressure, would discharge about 500 gal. per min. For delivery of 400 gal. per min. the loss of head would be about 15 ft., leaving a discharge head of 10 ft.; for 300 gal. a loss of head of about 8.5 and discharge

head of 16.5 ft.; for 200 gal. a loss of head of about 4 ft. with a discharge head of 21 ft.; and for 100 gal. per min. a loss of head of about 1 ft. with a discharge head of 24 ft.

QUESTION [Gallons per Inch Depth of Vertical Sided Tank]—*What number of gallons is contained per inch depth of an oil tank 10 ft. ½ in. in diameter with vertical sides, and what should be the diameter to contain 50 gallons per inch of depth?*
J. L. P.

Answer—One U. S. gallon is 231 cu. in., and the tank 10 ft. ½ in., or 120.5 in. in diameter would contain $120.5 \times 120.5 \times 0.7854 \div 231 = 49.37$ gal. per inch of depth. To contain 50 gal. per inch of depth the diameter would need to be $\sqrt{231 \times 50 \div 0.7854} = 121.26$ in., or practically 10 ft. 1¼ in.

QUESTION [Advantages of Turbo-Air Compressors] —*What are the advantages of turbo-air compressors?* A. W. L.

Answer—Compactness, adaptability to direct connection with high-speed steam turbines or electric motors and capability of yielding an output of compressed air free of oil or explosive hydrocarbon compounds that commonly result from cylinder lubrication of reciprocating air compressors.

QUESTION [Advantages of Two-Stage Air Compression]—*What are the advantages of two-stage compression of air over single-stage compression?* C. H. L.

Answer—Two-stage compression is adopted for the purpose of cooling the air and thereby obtaining a reduction of volume before the final compression. Intercooling, as by circulation of water, is an essential detail for obtaining any advantage from two-stage compression. The advantage over single-stage compression is an appreciable saving of power and the avoidance of the high temperatures, thus permitting of more satisfactory lubrication and less discharge in the air receiver and piping system to cause fires and explosions, and less drop of pressure from cooling of the compressed air.

Section V

Electricity

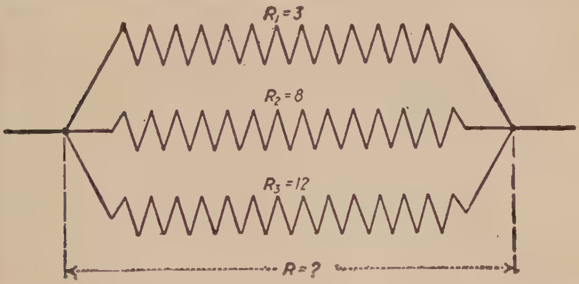
Section V

Electricity

Current, Voltage and Power Relations and Measurements

QUESTION [Joint Resistance of a Circuit]—*If three resistances, of* 3, 8 *and* 12 *ohms respectively, are connected in parallel, what is the joint resistance of the group?* R. G. S.

Answer—The joint resistance of a group of three resistances connected in parallel as in the figure, may be found from the



formula, $R = \dfrac{1}{\dfrac{1}{R_1}+\dfrac{1}{R_2}\div\dfrac{1}{R_3}}$ where R equals the joint resistance; and R_1, R_2 and R_3 the respective resistances.

In this problem $R = \dfrac{1}{\dfrac{1}{3}+\dfrac{1}{8}+\dfrac{1}{12}} = \dfrac{1}{\dfrac{13}{24}} = \dfrac{24}{13} + 1.846$ ohms.

QUESTION [Fall of Potential Method of Measuring Resistance]—*In a well-known book it is stated that a known resistance and voltmeter are required to make electrical resistance measurements by the "Fall of Potential Method." While talking with one who is an authority on the subject, I was told that a voltmeter and an ammeter are required to measure ohmic resistance of electric circuits by the "Fall of Potential Method." Which method is the "Fall of Potential Method"?* M. W. K.

Answer—Your question is one that is open to discussion. Although certain authors of books dealing with the subject use a voltmeter and a known resistance to measure the ohmic value of an unknown resistance and refer to it as the "Fall of Potential Method," in one of the electrical engineers' handbooks the statement is made, "The fall of potential method consists simply of noting the voltage drop with a known cur-

rent flowing through the resistance, and calculating the resistance from Ohm's law $R = \frac{E}{I}$." Since the volts drop is used in both cases, it is apparent that both schemes are "Fall of Potential Methods." In one case a known resistance is used with the voltage drop, and in the other the current is used with the volts drop to obtain the ohmic value of the unknown resistance.

QUESTION [Current-Capacity of No. 18 Copper Wire] —*What is the minimum length of No. 18 B. & S. copper wire that can be connected across a 110-volt circuit and have the temperature of the wire remain within a safe limit?* A. F.

Answer—A safe temperature limit is rather indefinite, since for one set of conditions it would be one thing and something else for another. If fibrous insulation is used on the wire, the temperature would have to be limited to about 200 deg. F., while with asbestos it could easily be double this value. If the wire were run in open air, it would carry a considerably higher current without overheating than when wound in a coil. If insulated with cotton and wound into a coil, the current should be limited to a value where the watts expended in the coil would not exceed the coil's outside area, in square inches, times 0.75. This value will vary with the shape of the coil. If the wire is run in open air, it will carry approximately 10 amperes continuously without overheating and would require a length of about 1,600 ft. to limit the current to this value on 110 volts.

QUESTION [Load Current and Voltage from Generator Voltage]—*If a load is supplied from a generating station over a transmission line, is there any way other than the cut-and-try method, of determining the voltage at the load and the current necessary to supply a certain number of kilowatts to that load, knowing the voltage at the generating station?* S W.

Answer—If the power supplied is direct current and the load is the only one supplied over that line, and if the resistance of the line is known, the current drawn for a certain load is given by the formula,

$$I = \frac{E - \sqrt{E^2 - 4W \times R}}{2R}$$

where I is the current required, E the voltage at the generating station, W the watts (1000 × kw.) consumed by the load, and R the resistance of the line in ohms. The load voltage is equal to $E - IR$.

QUESTION [Effects of Voltage Increase on Power Transmitted]—*State the effects on the current, resistance, diameter and weight of wire resulting in the transmission of a given power load by increasing the voltage four times, the percentage of loss remaining the same in both cases.* H. F. S.

Answer—Since the voltage is increased four times, the current will be decreased four times. The resistance of the conductors can be increased 16 times, which means that the cross-sectional area will be reduced by 16; consequently, the weight of the conductor will only be one-sixteenth that on the higher voltage that it would be on the lower, and the diameter of the conductor will be reduced 4 times. The foregoing is possible in practice only where the insulation of the conductor is sufficient to withstand the increased voltage and the conductor has sufficient mechanical strength to meet the requirements of operating conditions.

QUESTION [Kilovolt-Amperes and Power Factor]—*I am in charge of a 600-kv-a. alternating-current generator and would like to know the meaning of kilovolt-ampere and how horsepower is figured from it; also the meaning of power factor.* M. J.

Answer—The power (watts) delivered by a direct-current generator is equal to the volts times amperes. In an alternating-current generator this is true only when the current is in step with the voltage, consequently the product of the volts and amperes, with respect to power, may mean anything from zero to the value of their product; therefore kilovolt-amperes (kv.-a.) has been adopted as a unit for rating alternating-current generators, synchronous motors, synchronous condensers and transformers. For a single-phase circuit the kv.-a. $= \frac{EI}{1000}$; in a balanced two-phase circuit, kv.-a. $= \frac{2EI}{1000}$; and on a balanced three-phase circuit kv.-a. $= \frac{1.732EI}{1000}$, where E equals the voltmeter reading and I equals the ammeter reading. The wattmeter will give the load in kilowatts and this divided by kilovolt-amperes gives the power factor; that is, power factor $= \frac{\text{kilowatts}}{\text{kilovolt-amperes}}$; from which kilovolt-amperes $= \frac{\text{kilowatts}}{\text{power factor}}$ and kilowatts = kilovolt-amperes $\times$ power factor. Horsepower $= \frac{\text{kilowatts} \times 1000}{746}$ or $\frac{\text{kilovolt-amperes} \times \text{power factor} \times 1000}{746}$.

QUESTION [Calibrating Ammeter Shunts]—*In calibrating large shunts, is it possible to figure accurately the material to give a desired drop instead of filing or milling off a little at a time and taking a reading each time?* R. T.

Answer—After a shunt is built up and tested to determine the drop at its rated current, it is easy to determine how much metal must be milled or filed off to give the desired drop, especially if the shunt be composed of manganim plates, as is

usually the case. Assume a shunt in which the plates are exactly five inches wide. The length of the plates and their number is immaterial to this calculation. Suppose this shunt gives a drop of 95 millivolts at the rated current and it be desired to increase the drop to 100 millivolts. Divide the plate into 100 imaginary divisions; if all but one division be removed and the current forced through this one division, the drop will be 100 times greater, or 9,500 millivolts. If 95 of these imaginary divisions be left in parallel, the drop will be 95 times less than through one, that is, 100 millivolts. Therefore if it be desired to increase the drop from 95 millivolts to 100, we must cut off 5 per cent of the width of the shunt. In this particular case, the width must be cut down to 4.75 inches.

QUESTION [Calculating Power Factor]—*Our alternators are three-phase, 600-volt 1500-kw. machines. If the wattmeter indicates 1300 kw. and the ammeters each read 2000 amperes, what would the power factor be? Is low power factor the cause of poor voltage regulation?* E. Mc.

Answer—The kilovolt-amperes of a balanced three-phase circuit is kv.-a. $= \frac{EI \times 1.732}{1000}$, where E is the volts between terminals and I the current in amperes per terminal. Then in this problem kv.-a. $= \frac{600 \times 2000 \times 1.732}{1000} = 2018.4$. The power factor (p.f.) equals kilowatts (kw.) divided by the kilovolt-amperes (kv.-a.), or p.f. = kw. ÷ kv.-a. = 1300 ÷ 2078.4 = 0.62. Low power factor is one of the causes of poor voltage regulation. A power factor of 0.62 is considerably lower than would be considered good operating practice.

QUESTION [Effects of Low Power Factor]—*Will an alternating-current generator be affected by poor power factor, when the generator is supplying power directly to lightly loaded induction motors, the same as when the load is supplied through transformers?* E. G. H.

Answer—It will not make any difference whether the generator supplies the power directly to the motors or through transformers to the motors, the kilowatt capacity of the generator will be reduced in either case by the low power factor of the motors. However, the power factor of the system will be lower where the transformers are used, on account of the power factor of the transformer being less than unity even under full load.

QUESTION [Reactive Factor]—*What is the reactive factor of an alternating-current system, how is it determined, and what relation has it to the power factor?* M. O. W.

Answer—The reactive factor of an alternating-current circuit is the sine of the angle of lag or lead between the current and voltage. If the power factor is read from an instrument or calculated by dividing the watts by the volt-amperes, refer-

ence can be made to a table of natural sines and cosines and find the sine of an angle corresponding to the cosine that is the power factor. For example, if the power factor is 0.86, this is the cosine of a 30-deg. angle and the sine of 30 deg. is 0.5, therefore the reactive factor is 0.5 when the current is out of step with the volts by 30 degrees. When the current and volts are in phase, the power factor is 1.00 where the reactive factor is 0. Another way of determining the reactive factor is by the formula:

$$\textit{Reactive factor} = \sqrt{1 - \left(\frac{\textit{Watts}}{\textit{Volt-Amperes}}\right)^2}$$

The volt-amperes times the power factor equals true watts, where the volt-amperes times reactive factor equals the reactive component of the volt-amperes, sometimes called magnetizing watts, that is being transmitted back and forth through the circuit doing no useful work.

QUESTION [Relations of Currents in A. C. Systems] —*For transmitting a like amount of power at the same voltage and power factor, what would be the value of the current in each of the following systems:* (*a*) *single-phase,* (*b*) *four-wire two-phase,* (*c*) *three-wire two-phase,* (*d*) *three-wire three-phase, and* (*e*) *four-wire three-phase?* W. D. T.

Answer—If I is the current in amperes, W the power in watts, E the voltage, and $P.F.$ the power factor, we would have (a) $I_a = \frac{W}{E \times P.F.}$; (b) $I_b = \frac{\hat{I}_a}{2}$; (c) in the two outside wires $I_c = I_b$ and in the common wire $I_c = \sqrt{2\,I_b}$; (d) $I_d = \frac{I_a}{\sqrt{3}}$; and (e) in the three outside wires $I_e = \frac{I_a}{\sqrt{3}}$; and in the common wire $I_c, = 0$. The foregoing conditions are true on the assumption that load is evenly balanced between the phases.

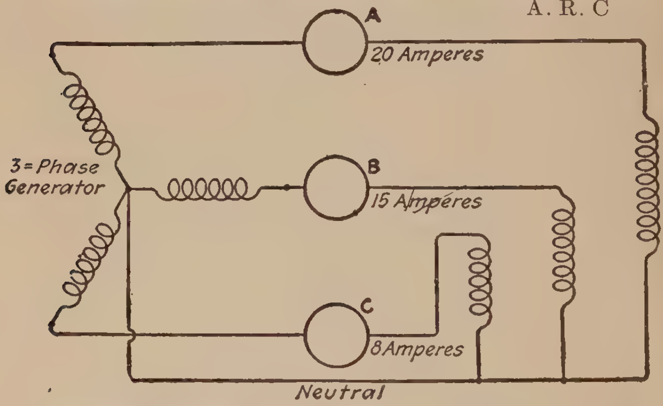
QUESTION [Watts in a Three-Phase Circuit]—*In testing three-phase induction motors, we connect a single-phase indicating wattmeter in one phase, and then in one of the other phases? the sum of the two readings gives the total watts consumed. If all three phases are balanced, why could we not get the total watts by taking a reading in one phase only and multiplying it by* 1.732? E. C. H.

Answer—Your statement that the sum of the wattmeter readings gives the total load in watts is only correct when the power factor is greater than 0.50. When the power factor is less than 0.50, the difference of the two readings gives the total watts. If the power factor is unity, the readings of a wattmeter connected in one phase and then in another of a

balanced three-phase circuit would be the same, and the total power would be the sum of the two readings, or the reading of one multiplied by 2 and not 1.732, as suggested in the question. However, this can only be true for a balanced circuit having a constant load at unity power factor, since in every other case the two readings will not be the same value, and as stated in the foregoing must be added when the power factor is greater than 0.50 and subtracted when less. The constant 1.732 is used only to obtain the apparent watts, or volt-amperes, in a three-phase circuit, which when circuit is balanced is $W = EI \times 1.732$ where W equals the total volt-amperes transmitted, E volts between terminals and I amperes per terminal.

QUESTION [Current in the Neutral of a Three-Phase System]—*Three ammeters, A, B and C, are connected in a four-wire three-phase system and read 20, 15 and 8 amperes respectively, as shown in the figure. What is the current in the neutral conductor if the power factor of each load is unity?*

A. R. C



Answer—When the power factor of the system is unity, the current in the neutral of a four-wire three-phase system is always equal to

$$I_N = \sqrt{(I_A - I_B)^2 + (I_B - I_C)^2 + 2\,(I_A - I_B) \times (I_B - I_C) \times \cos 60 \text{ deg.}}$$

where I_A, the reading of ammeter A, equals 20 amperes; I_B, the reading of ammeter B, equals 15 amperes; and I_C, the reading of ammeter C, equals 8 amperes.

Then,

$$I_N = \sqrt{(20-15)^2 + (15-8)^2 + 2\,(20-15) \times (15-8) \times 0.5}$$
$$= \sqrt{5^2 + 7^2 + (2 \times 5 \times 7 \times 0.5)} = 10.44 \text{ amperes.}$$

It should be remembered that to use the arrangement of symbols in the formula it is necessary that I_A have the highest value, I_B the intermediate value and I_C the smallest value.

QUESTION [**Current per Terminal in a Three-Phase Circuit**]—*Each of the lamps in Fig. 1 takes 5 amperes, which is in phase with the voltage applied to it. What is the value of the*

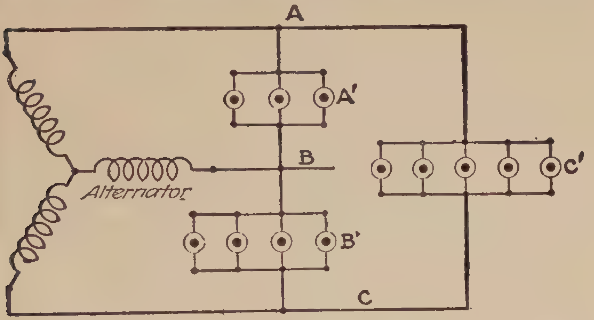
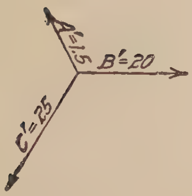
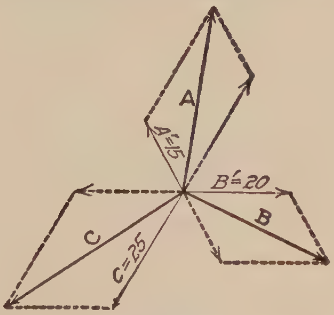


FIG. 1. DIAGRAM OF ALTERNATOR AND LOADS

current flowing in each line wire? The generator is connected in star and gives a correct three-phase voltage. E. C. B.

Answer—Since each lamp takes 5 amperes, load **A′** will take $3 \times 5 = 15$ amperes, load B', $4 \times 5 = 20$ amperes and load **C′**, $5 \times 5 = 25$ amperes. The different currents are in step with the voltage across their respective circuits, therefore they are 120 deg. apart and may be shown vectorially as in Fig. 2. The current in line A is the vectorial sum of loads **A′** and **C′**, that in line **B** the vectorial sum of loads **A′** and **B′** and that in line C the vectorial sum of loads B' and C', flowing toward or away from the respective conductors. Then by combining the loads in Fig. 2 vectorially, as shown in Fig. 3,





the resultants A, B and C will represent to scale, respectively, the value of the current in A, B and C, Fig. 1. In Fig. 3

$$A = \sqrt{(A')^2 + (C')^2 + 2A'C' \cos 60 \text{ deg.}}$$

$$B = \sqrt{(A')^2 + (B')^2 + 2A'B' \cos 60 \text{ deg.}}$$

$$\text{and } C = \sqrt{(B')^2 + (C')^2 + 2B'C' \cos 60 \text{ deg.}}$$

As cos 60 deg. = 0.5, then,

$$A = \sqrt{15^2 + 25' + 2 \times 15 \times 25 \times 0.5} = 35 \quad \textit{amperes}$$
$$B = \sqrt{15^2 + 20^2 + 2 \times 15 \times 20 \times 0.5} = 30.4 \textit{ amperes}$$
$$C = \sqrt{20^2 + 25^2 + 2 \times 20 \times 25 \times 0.5} = 39 \quad \textit{amperes}$$

That is, the current in A conductor, Fig. 1, is 35 amperes, in B conductor 30.4 amperes and in C conductor 39 amperes.

QUESTION [Volts Drop in Power Line]—*If* 250,000-*circ.mil. conductors were used to transmit* 800 *hp., having a power factor of* 0.80, 8000 *ft., at* 2200 *volts, on a three-phase* 60-*cycle system, with the conductors spaced in a triangle* 24 *in. apart, what would be the volts drop in the line?* G. G.

Answer—The volts drop in a three-phase circuit maybe calculated in the same way as for a single-phase circuit transmitting one-half the load. Therefore, in this problem, find one-half the three-phase load in amperes and proceed as if calculating a single-phase circuit. One half the single-phase watts W = hp. × 746 ÷ 2 = 800 × 746 ÷ 2 = 298,400 and the current I,

$$I = \frac{W}{E \times P.F} = \frac{298{,}400}{2200 \times 0.80} = 170 \text{ amperes.}$$

The volts drop in the circuit is made up of two parts, one the resistance component and another the reactance component. If E_R represents the volts drop due to the resistance of the conductors and E_X the volts drop due to the reactance of the circuit, we have

$$E_R = \frac{2LIR}{1000}$$

where L equals length of conductors in feet one way and R the resistance per 1000 ft. of conductor; and

$$E_X = \frac{2LIX}{1000}$$

where X equals the reactance drop in volts per ampere per 1000 ft. of conductor at 60 cycles, conductors space 24 in. apart; as obtained from a table $X = 0.110$. Then

$$E_R = \frac{2 \times 8000 \times 170 \times 0.042}{1000} = 114.24 \text{ volts}$$

and

$$E_X = \frac{2 \times 8000 \times 170 \times 0.110}{1000} = 299.2 \text{ volts}$$

The volts drop in the line is

$$E_d = \sqrt{E^2_R + E^2_X} = \sqrt{114.24^2 + 299.2^2} = 320.3 \text{ volts}$$

This is equivalent to 320.3 ÷ 2200 = 14.55 per cent. drop, which is considerably higher than would be considered good practice.

QUESTION [Size of Conductors]—*What size of conductors will be required to transmit* 1600 *hp., at* 0.75 *power*

factor, 15 miles with a loss of 7 per cent. of the power generated in the line; the circuit to be three-phase with 11,000 volts at the generator, conductors spaced 36 in. apart? Also, what will be the per cent regulation? R. M. S.

Answer—With 11,000 volts at the generator the voltage to neutral of a three-phase circuit will be $11{,}000 \div 1.732 = 6350$. Assuming 15 per cent. regulation, the volts E to neutral at the receiver end of the line will be $6350 \div 1.15 = 5520$. The watts $W = \text{hp.} \times 746 = 1600 \times 7.46 = 1{,}193{,}600$, and the current, I, per wire, when the power factor is 0.75, is $I = W \div 3E \times P.F. = 1{,}193{,}600 \div (3 \times 5520 \times 0.75) = 96$ amperes. Allowing the loss in the line to be 7 per cent. of the power generated, the watts loss per wire is $W_n = \frac{W}{3} \times \frac{0.07}{0.93} = \frac{1{,}193{,}600}{3} \times \frac{0.07}{0.93} = 29{,}947$. Resistance per wire $= W_n \div I^2 = 29{,}947 \div 96 \times 96 = 3.25$ ohms; and the resistance per mile is $3.25 \div 15 = 0.217$ ohm. The nearest size standard wire is a 250,000-circ.mil conductor, which has a resistance of 0.2253 ohm, making the total resistance of one wire equal $0.2253 \times 15 = 3.38$ ohms. With conductors spaced 36 in. apart, the inductive reactance in ohms, at 60 cycles, per mile of single 250,000-circ. mil conductor is 0.626, making a total of $0.626 \times 15 = 9.39$ ohms.

Knowing the receiver voltage E, the generator voltage Eg is found by the formula,

$$Eg = \sqrt{(E \cos \Theta + IR)^2 + (E \sin \Theta + IX)^2}$$

where R is the resistance ohms and X the reactive ohms of one conductor, and Θ the angle corresponding to the power factor in the circuit. For a power factor of 75 per cent. $\cos \Theta = 0.75$ and $\sin \Theta = 0.66$. Then

$$Eg = \sqrt{(5520 + 0.75 + 96 \times 3.38)^2 + (5520 \times 0.66 + 96 \times 9.39)^2}$$
$$= 6370 \text{ volts}$$

$$\textit{Per cent. regulation} = \frac{Eg - E}{E} \times 100 = \frac{6350 - 5520}{5520} \times 100 = 15 \textit{ per cent.}$$

This value is close enough for all practical purposes. It should be noted that there is no direct method of making these calculations. A certain per cent. regulation is assumed and the calculations made. If this does not come close enough, another value is assumed and the calculation made again until the correct value is obtained.

QUESTION [Measurement of Three-phase Power]—*How can the power consumption of a three-phase induction motor be measured with an ammeter and a voltmeter?* B. S. M.

Answer—It is not possible to measure power in an alternating-current motor circuit with an ammeter and a voltmeter unless the power factor is known; for the power factor changes with the load, with the result that the current is not proportional to the load. For example, the motor in question might take 40 amperes at full load, 32.5 amperes at three-

quarter load, 25 amperes at one-half load, 18 amperes at one-quarter load, and 14 amperes at no load. Therefore reduction of load from full to three-quarters decreases the current by 7.5 amperes, whereas from one-quarter to no load decreases the current only 4 amperes.

QUESTION [Polyphase Wattmeter Reading—*A switchboard polyphase watt-hour meter is connected through two potential and two current transformers on a 2,300-volt three-phase circuit. If the potential transformer and current transformer connected to one of the meter elements are removed from the circuit, what percentage of the load would the other meter element register?* W. T. L.

Answer—When a polyphase watt-hour meter is correctly calibrated and connected to a balanced three-phase circuit operating at unity power factor, each element will register 50 per cent of the power transmitted. If the power factor is less than unity, or the load is unbalanced, then one element will record more than 50 per cent and the other will record less, but the sum of the two in all cases will equal 100 per cent. From the foregoing it is evident that if one of the meter elements is disconnected and the load is balanced at unity power factor, the other element will record 50 per cent of the power transmitted. But under any other condition the registration of one element will depend upon the circuit conditions.

QUESTION [Size of Instrument Transformer]—*How are the sizes of potential and current transformers for a polyphase meter determined? For instance, we have a polyphase wattmeter rated at 60 cycles, 80 amperes, 110 volts, connected to a three-phase 2,300 volt supply circuit through potential and current transformers. The potential transformers are marked "20:1 ratio, 0.05 kva." and the current transformers are marked "2,500 to 4,500 volts 16:1."* V. K. S.

Answer—Current transformers are, in general, designed to give 5 amperes on the secondary at full load, and all ammeters, wattmeters, power-factor meters, etc., are designed, when supplied from current transformers, for 5 amperes in the current coil. Potential transformers are usually designed to step the voltage down to 110 volts, the transformers themselves to have 0.05 kilovolt-amperes capacity. This is standard practice, although sometimes there are exceptions made to this rule, in special cases.

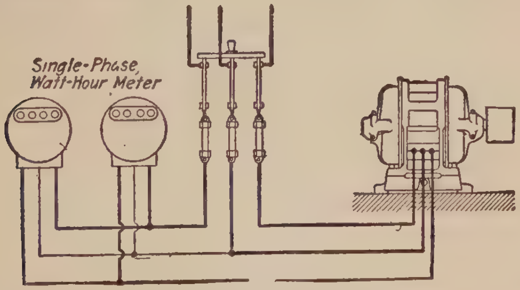
QUESTION [Wattmeters Require Adjustment]—*On one of the feeders going out from our plant the voltage is stepped up from 2,300 to 13,200 and, at the end of a five-mile transmission line, stepped down again to 2,300 volts. The power is metered on the 2,300-volt side of the transformers at both ends of the line. Should not both meters register the same? As it is now, the meter at the out end of the line registers considerably more than the one at the power plant.* V. A. S.

Answer—The two meters should not register the same; however, they are now doing the reverse of what is the correct condition. The instrument at the load end of the line should register less than the one at the power house, since the latter meters not only the power supplied to the load, but also the losses in the transformers and transmission lines, whereas the meter at the load measures only that which is supplied to the load. If both meters are correctly calibrated, the one at the load will register approximately 10 to 15 per cent less than the one at the plant. The ratios and connections of the instrument transformers should be checked to make sure they are correct, and then the meters should be calibrated.

QUESTION [**Metering Three-Phase Power**]—*I have a* 75-*hp*. 220-*volt three-phase* 60-*cycle motor and wish to install integrating wattmeters. As the meters on hand were rated at only* 150 *amperes, I put two of them in parallel on one phase, as shown in the figure. With this connection the meters ran very slowly when the motor was loaded; when the load was light the meters ran backward. What caused this action of the meters?*

C. S. H.

Answer—The action of the meters was due to their being connected in the low-reading side of the circuit. If a watt-hour meter is connected first in one phase of a three-phase circuit and then in another phase, it will be found that the readings will not be alike unless the load is balanced and the power factor 100 per cent. When the power factor is less than 100 per cent., even if the load is balanced, as it generally is in the case of an induction motor, the meter will read higher on one phase than it does on the other as ong as



the power factor is greater than 0.50. When the power factor is less than 0.50, if the meter is connected to one phase it will read fast and will reverse when connected to the opposite phase. This is what happened as the meters were connected in the low-reading phase. When the motor was loaded, the power factor was above 0.50 and the meters registered slowly in the proper direction, but when the load was removed from the motor the power factor dropped below 0.50 and the meters reversed and ran in the opposite

direction, as they should. If the meters had been connected in the opposite phase, they would have run fast and indicated an excess load on the motor. The only way that three-phase power can be metered satisfactorily is with two single-phase meters connected into two of the phases, or with a polyphase meter, which is practically nothing more than two single-phase meters combined in one instrument.

QUESTION [Increasing Length of Meter Connections]—*In connecting up alternating-current meters on the switchboard, does it affect the accuracy of the meter to splice four or five feet of wire to the potential- and current-transformer secondary leads so as to make them reach the meters?* V. K. S.

Answer—The only case where the accuracy of electrical measuring instruments is affected by a change in the length of the leads is when the instrument is connected to a shunt, such as direct-current ammeters and in some cases the current coils of watt-meters. With alternating-current instruments connected to the secondaries of transformers the length of the leads may be changed without affecting the accuracy of the instrument.

Generators: Direct-Current, Alternators and Induction

QUESTION [Sparking with Too Strong Interpoles]—*What would be the effect of too many turns on the interpoles of a direct-current machine?* R. A.

Answer—The object of the interpole is to produce a magnetic field which causes the current in the coil under commutation to reverse and build up in the opposite direction. If the field is too weak, the reversal will occur too late and there will be sparking; if it is too strong, the reversal will come too early, the current will build up to too high a value, and will have to be suddenly reduced as the commutator bar passes out from under the brush, and there will also be sparking.

QUESTION [Rating of Alternating-Current Generators]—*What is the kilowatt rating of a 550-volt 300-amperes three-phase alternator at 0.80 power factor? What would be the machine's kilowatt rating at unity power factor?* H. A L.

Answer—At unity power factor the alternator would have a kilowatt rating equal to volts × amperes × 1.732 × power factor ÷ 1000= 550 × 300 × 1.732 × 1 ÷ 1000 = 285.78. This means that this machine when operated at unity power factor can supply a load of 286.78 kw. At 0.80 power factor the kilowatt rating= 550 × 300 × 1.732 × 0.80 ÷ 1000= 228.62; that is, at 80 per cent. power factor the alternator can supply only a 228.62 kilowatts load.

QUESTION [Rotation Direction of Alternator]—*Does it make any difference in which direction a newly installed alternator revolves? Or will the exciter connections to the alter-*

nator have to be reversed in case the generator is rotated in a direction opposite from that intended by the manufacturer? The exciter is driven from the alternator. E. G. H.

Answer—If the machine is operated as a single unit, it will not make any difference in which direction the rotor revolves. If the rotation of the exciter is reversed, it will be necessary to cross either its armature or field leads so that the machine will generate. Where the alternator is operated in parallel with other machines care will have to be taken to see that the new machine has the proper phase relation to the bus voltage.

QUESTION [Voltage Drop in Field Winding and Rheostat]—*In a direct-current generator how does the voltage drop in the field rheostat compare with that in the field winding?* F. M

Answer—If the generator is shunt-wound, it is customary to design the field rheostat to absorb about one-half of the line voltage, while in compound-wound generators this is usually reduced to about one-third of line voltage. The voltage drop in the rheostat will therefore be equal to that in the field winding in the first case and to one-half of the drop in the field in the second case.

QUESTION [Parallel Operation of Alternators]—*When two alternators are operating in parallel, why is it impossible to regulate the load on each machine by varying their field excitation, as in the case of direct-current machines?* D. R. A.

Answer—The only way to control the load distribution between two alternating-current generators operating in parallel, is to adjust the governor as if to vary the speed of the prime mover; for any attempt to raise the voltage on one machine by increasing its excitation will simply set up a circulating wattless current between the two machines, which will weaken the field in the overexcited machine and strengthen it in the other, resulting in a slight increase in the line voltage, but practically no change in the load distribution.

QUESTION [Load Distribution of Alternators in Parallel]—*Do alternators in parallel change their speed, consequently the frequency, in changing their load distribution, or do they change only the relative position of the main field with respect to the armature field?* H. G. S.

Answer—When changing the load distribution between them, alternators in parallel often change both their speed —consequently the frequency—and the relative magnetic positions of main and armature fields; but their relative speed remains the same, since they must remain in step. Thus, if it is desired to increase the proportion of load taken by one machine, the prime mover driving that machine is

speeded up, the rotor of the generator jumps forward while that of the other generator drops back, with respect to their respective armature fields, and the first machine takes some of the load off the second, pulling it up to the new speed. The speed and frequency of the two machines have therefore increased. They may be kept constant, however, by a double adjustment of speed, raising it slightly on one machine and lowering it on the other.

QUESTION [Efficiency of Induction and Synchronous Generators]—*When a synchronous alternator and an induction generator operate in parallel, which is the more efficient?* F. G. M.

Answer—If the two machines are of about the same capacity, the induction generator probably will show 1 to 2 per cent less efficiency than the synchronous machine.

QUESTION [Operation of an Induction Generator]—*Will you explain the operation of a squirrel-cage type induction motor when operating as a generator as sometimes used in semiautomatic hydro-electric power plants?* J. T.

Answer—When a squirrel-cage type induction machine is operating as a motor, its rotor is running slower than the revolving magnetic field and is cutting this field in a direction to set up a current in the rotor of a direction to produce a torque in the same direction as the field is moving. This rotor current is in an opposite direction to the stator current, consequently when operating as a motor the rotor current is a demagnetizing current, which will tend to decrease the strength of the magnetic field and therefore the value of the counter-electromotive force in the stator.

If power is applied to the rotor, as in the case of the induction generator, and driven faster than the revolving magnetic field—that is, above synchronism—its conductors will be cutting the lines of force in a direction opposite to that when running as a motor. Therefore, if when operating as a motor, the current in the rotor is a demagnetizing current, and tends to reduce the counter-electromotive force in the stator winding, then when operating as an induction generator, the current in the rotor will be a magnetizing current and will increase the flux, causing the counter-electromotive force to increase above the applied voltage, and the machine will become a generator supplying current to the circuit to which it is connected instead of taking power.

QUESTION [Grounding in a Three-phase Generating Plant]—*Is it good practice to ground the steel framing of the plant, as well as the instrument casings and transformer tanks, etc., on the same ground plate as the generator neutrals?* C. R.

Answer—This is permissible and is the practice in some plants. But it is considered better practice to have a separate

ground for the generators. No matter how much care is taken to keep the resistance of the generator ground as low as possible, it is practically impossible to prevent the voltage drop in the ground from reaching at least 200 volts on dead short-circuit. In case the same ground is used for the station, this voltage backs up all over the plant, and anyone touching any apparatus may receive a shock, and delicate instruments be damaged by this tension. Furthermore, there is always some chance, although remote, that the ground may accidentally dry out or the connection to the plate may become corroded and broken, in which case a short-circuit in one phase may give full-line voltage on the whole grounding system of the plant, with great danger to life and apparatus

QUESTION [Induction Generator for Small Water Power Plants]—*What would be the advantages of an induction generator for a small water-power plant tied into a large electric system?* S. V. B.

Answer—The advantages would be simplicity and sturdiness of construction, small care for operation, requiring little more attention than necessary lubrication, no requirement of exciter, automatic adaptation of electric load to the variable water power available, and no requirement of a waterwheel governor.

Direct-Current Motors

QUESTION [Loss in Direct-Current Motor at Different Speeds]—*Is there any loss of power if the speed of a direct-current motor is changed from normal speed, by introucing resistance into either the field or armature circuit?* E. N.

Answer—The power loss at different speeds in an adjustable-speed direct-current motor depends on the method of speed control. If the speed is controlled by varying the field strength, the efficiency will be approximately the same at all speeds, provided the current drawn by the machine remains constant. If the speed is regulated by means of a rheostat in the armature circuit, the power loss is equal to that absorbed in the rheostat. If the current is kept constant, the power lost is directly proportional to the voltage drop in the rheostat, and therefore to the drop in speed, since the latter is proportional to the voltage drop in the resistance.

QUESTION [Varying Speed of a Compound Motor]—*Can the speed of a compound-wound motor be adjusted by the use of a variable resistance connected in series with the shunt-field winding as in shunt-wound motor?* W. N. C.

Answer—The speed of a compound motor cannot be adjusted by introducing a variable resistance in the shunt-field winding, and operate as a compound machine, as with a shunt motor. This is due to the ampere-turns of the series winding becoming more effective as the ampere-turns of the shunt winding are reduced. Where a drive requires a com-

pound motor during the starting period and shunt-field control after, the series winding is cut out of circuit before introducing resistance into the shunt-field circuit. Such conditions are found in elevator and printing-press drives, etc.

QUESTION [Reversing a Compound-Wound Interpole Motor]—*We have a 40-hp. 220-volt compound-wound interpole motor with the armature, series field and interpole windings connected in series and do not see any provision made for reversing the direction of rotation of this machine. What connection should be changed to give a reverse direction of rotation?* W. L.

Answer—The direction of rotation of any interpole motor may be changed by reversing the direction of the current through both the armature and interpole windings. If

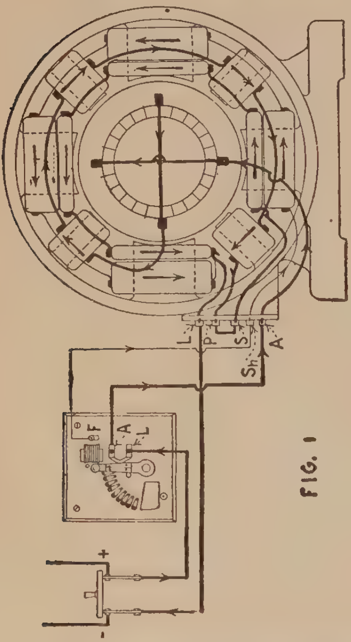


FIG. 1

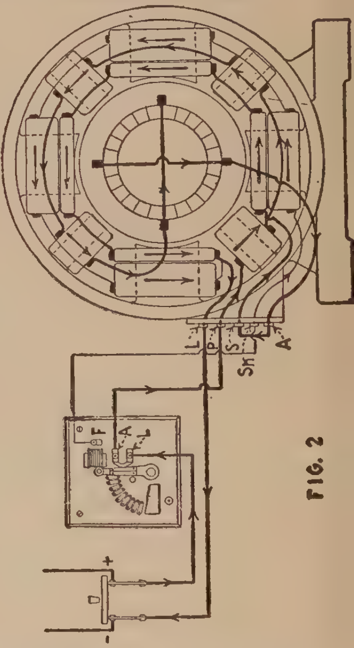


FIG. 2

Fig. 1 is the correct connection to give a compound-wound interpole motor one direction of rotation, the connections Fig. 2 are correct for the opposite direction of rotation. It will be seen that the current in the armature and interpole windings in Fig. 2 is flowing in an opposite direction from that in Fig. 1, which was accomplished by interchanging the external wires to the interpole terminal *P* and the armature terminal *A*. If the machine is shunt-wound, it is generally more convenient to interchange the shunt-field connections to obtain a reverse direction.

QUESTION [Compound-Wound Motor Reverses When Starting]—*What would cause a compound-wound motor to reverse its direction of rotation upon reaching the third point of the starting rheostat? What would cause the same motor to run within normal speed limits at one starting and accelerate to a dangerous speed at another starting?*

H. W. C.

Answer—The trouble is caused by the series and shunt-field windings being connected in opposition. The starting resistance was cut out at a rate that allowed the armature current to build up to a value, when the third point of the starter was reached, that made the ampere-turns of the series winding greater than the shunt-winding ampere-turns. This reversed the polarity of the polepieces and consequently reversed the direction of the motor. In general, if a compound-wound motor with its field winding connected in opposition is starting under light load and is accelerated slowly, it will come up to normal speed without giving any evidence of anything being wrong. However, if an attempt is made to accelerate the same motor rapidly, it will reverse before the starting resistance is cut out, as explained in the foregoing, or it may run at a dangerous speed due to the demagnetizing effect of the series-field winding.

QUESTION [Keeping Motor in Service]—*How can a shunt- or compound-wound motor be started and kept in service if the starting box burned out?*

E. P.

Answer—The connections for starting a shunt and a compound motor with a water rheostat are shown in Figs. 1 and 2 respectively. The water rheostat is made from a bucket of water with a handful of common salt in it. It is preferable that a wooden bucket be used. One metal electrode is placed in the bottom of the bucket, and the other is gradually lowered into the solution at starting until the two

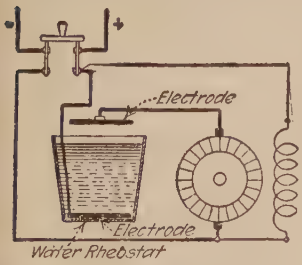


FIG. 1. CONNECTIONS FOR STARTING SHUNT MOTOR WITH WATER RHEOSTAT

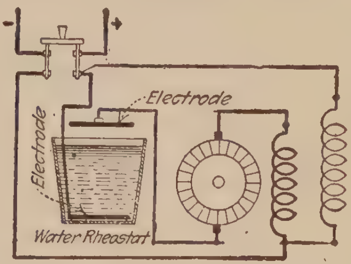


FIG. 2. CONNECTIONS FOR STARTING COMPOUND MOTOR WITH WATER RHEOSTAT

electrodes are together. When the motor is shut down, care should be exercised to see that the top electrode is removed from the bucket, or the fuses will be blown if the switch is closed with the electrodes together.

QUESTION [Carbon Brush Wear]—*In one of our direct-current motors that has four brushes on the commutator, two of the brushes wear away faster than the other two. What would cause this uneven wear when all four brushes are of the same grade?* T. A.

Answer—The brushes from which the current flows into the commutator always wear away faster than those through which the flow is from the commutator. The brushes through which the current flows into the commutator would be positive on a motor and negative on a generator. The additional wear on these brushes is due to small particles of carbon carried by the current from the brush contact surface and deposited on the commutator. If two of four brushes, of the same polarity, were to wear away faster than the other two, a difference in resistance in the shunts from the brushes to the brush-holder, or a difference in the tension on the brushes, would cause an unequal distribution of the current between the brushes and unequal wear.

Alternating-Current Motors—Synchronous and Induction

QUESTION [Size of Synchronous Motor]—*In our plant the present maximum load amounts to about* 325 *kw. At this load the power factor is around* 0.72. *We are putting in some additional equipment to be driven from a lineshaft, and from what can be determined from some of our other drives,* 78 *hp. will be required to drive the new installations. I would like to know the size of synchronous motor to install to drive the load and also raise the power factor to* 0-90. M. A. B.

Answer—Synchronous motors usually are rated in kilovolt-amperes, therefore in this problem the motor will be rated in kva. The mechanical load on the motor will be 78 *hp.* or $(78 \times 746) \div 1{,}000 = 58$ kilowatts. This makes the total kilowatt load equal to $325 + 58 = 383$. The kilo-volt-ampere load equals kilowatts divided by power factor; therefore, in the first case kva. $= 325 \div 0.72 = 452$, and in the second, kva. $= 383 \div 0.90 = 426$. Wattless component, $WC = \sqrt{\text{kvs.}^2 - \text{kw.}^2}$; then, with the present load $WC = \sqrt{452^2 - 325^2} = 314$, and with the synchronous motor added $WC' = \sqrt{426^2 - 383^2} = 186$ kva The difference between WC and WC', or $314 - 186 = 128$ kva., represents the leading wattless component that must be supplied by the synchronous motor, in addition to doing 58 kw. of mechanical work. Then the total motor capacity required is equal to the square root of the sum of the squares of the two loads, or $\sqrt{128^2 + 58^2} = 140$ kva., and the power factor of the motor equals kilowatts divided by kilovolt-amperes $= 58 \div 140$

= 0.41. It will be seen from the foregoing that by installing a 140-kva. synchronous motor and operating it at 0.41 power factor leading, it will do 78 hp. of mechanical work and raise the power factor of the total load to 0.90, and the kilovolt-ampere load on the generators will be reduced from 452 to 426 kilovolt-amperes; besides, under the new condition the voltage regulation will be improved.

QUESTION [Synchronous Motor for Power-Factor Correction]—*When synchronous motors are used for power-factor correction, what horsepower capacity should be used per 100-hp. capacity of induction motors? The sizes of the induction motors range from 10 to 30 hp., and the system has about 200-hp. capacity.* E. G. H.

Answer—There is no specified number of synchronous-motor horsepower capacity required for power-factor correction per 100 hp. of induction motors installed. The synchronous-motor capacity required will depend upon how much of the mechanical load is to be driven by this motor, the value of the power factor before correction and the amount of the correction. If the induction motors are lightly loaded, the power factor will be low and a greater kilovolt-ampere capacity will be required in the synchronous motor to correct the power factor than where the induction motors are operating near their rated capacity and the power factor is at a comparatively high value.

QUESTION [Power Input to Induction Motor]—*How is a 10-hp. three-phase 440-volt induction motor tested to determine if it is underloaded? Can an ammeter be used for testing?* P. M.

Answer—An ammeter reading of an induction-motor load is of little import unless the power factor is known. Since the power factor of an induction motor decreases as the load is reduced, the current will not be reduced in proportion to the load. An induction motor operating at 50 per cent load will take approximately 80 per cent of the full-load current, but the power factor will be in the neighborhood of 0.70, where at full load it is about 0.87. The most satisfactory way to measure the power input to an induction motor is with a polyphase wattmeter, and then allow 900 watts per horsepower output, to obtain the mechanical load on the motor. About the only thing that an ammeter would show is whether the motor is overloaded or underloaded.

QUESTION [Current Taken by Induction Motor]—*How is the current taken by an induction motor calculated? I have a 25-hp. three-phase 60-cycle 220-volt induction motor and would like to know how to figure the full-load current.* V. K. S.

Answer—The current per terminal taken by an induction motor usually can be found on the name-plate of the machine. The following formulas give values close enough for practical

purposes: For single-phase motors the full-load current per terminal is: amperes $= \frac{\text{horsepower} \times 1000;}{\text{volts}}$ for 4-wire, two-phase motors, amperes per terminal $= \frac{\text{horsepower} \times 1000}{\text{volts} \times 2}$ for 3-wire operation the current in the outside wires is the same as for a 4-wire circuit, but the current in the center leg is equal to the outside-leg amperes times 1.414. For a three-phase motor the amperes per terminal $= \frac{\text{horsepower} \times 1000,}{\text{volts} \times 1.732}$ or in your problem equal $\frac{25 \times 1000}{220 \times 1.732} = 65.6$ amperes.

QUESTION [Starting an Induction Motor]—*A 3-hp. 220-volt three-phase induction motor, driving a shaper, takes 60 amperes for about 20 seconds when first started. The current then drops to 5 amperes. Is this surge of current liable to harm the motor, and is the use of a starting compensator warranted?*

A. E. W.

Answer—A starting current of 60 amperes is not abnormal for the type of motor in question. And unless there is danger of the motor injuring the machine that it is starting, a starting compensator is not necessary. If you wish to properly protect the motor, you will have to do it by installing a double-throw switch with two sets of fuses—one set of about 15-ampere capacity on the running side, and another on the starting side of 60-ampere capacity.

QUESTION [Short-Circuited Wound Rotor]—*We have a 150-hp. 2200-volt three-phase 25-cycle motor. The rotor is wound and is rated at 263 volts and 245 amperes. Would it be possible to short-circuit the rotor leads and start the motor under about one-half rated load, with a compensation?* P. F. M.

Answer—With the slip rings short-circuited the rotor may develop considerable starting torque if the winding has a comparatively high resistance. However, at best it is liable to develop locking points and, if the resistance of the rotor's winding is low, may give trouble to start a load requiring only a small starting torque. If it is necessary to use a compensator to start the motor with, instead of a resistance in the rotor circuit, better results could be obtained by short-circuiting the rotor conductors on rings at each end of the core, thus converting the machine into one of the squirrel-cage type. However, at best this is only a makeshift and is liable to give trouble. If satisfactory operation is to be expected of a motor of this size and voltage, it should be started in the way it was designed to be; namely, with an external rotor resistance.

QUESTION [Motor Wire Not All the Same Size]—*We have a 20-hp. induction motor operating on a 230-volt three-phase 60-cycle circuit. The motor is connected to the starting device*

with two No. 3 *and one No.* 0 *conductor. What effect if any has the difference in size of conductors on the operation of the motor?* W. P. B.

Answer—It is doubtful if the machine in question is a three-phase motor, but more likely a 3-wire two-phase machine, although the former may be correct. In 3-wire two-phase systems the cross-section of the center wire is usually made 1.414 times that of one of the outside wires; this to take care of the current in the center leg of the system, which is 1.414 times that in one of the outside legs. However, if your machine is a three-phase motor, the one larger wire will have little effect on the operation of the motor. It would have a tendency to unbalance the voltage of the three phases, but this will be so slight as to be practically negligible.

QUESTION [**Changing the Speed of a Fan Motor**]—*Is it possible to increase the speed of direct-current motors used to drive fans? If so how is it done?* A. R.

Answer—If a fan is driven by a series motor, the speed of the motor may be varied by connecting a resistance in parallel with the field winding. If the motor is of shunt type, its speed may be varied by connecting a resistance in series with the field winding. The value of the resistance will depend upon how much the speed is increased and upon the size of the motor. It is doubtful if a motor that has been designed to drive a fan of a given size can have its speed increased very much without being overloaded, since a fan load increases as the cube of the speed.

QUESTION [**Fuse Blown on Three-Phase Motor**]—*What would be the effects of a fuse blowing in either of the primaries or secondaries of three transformers connected in star, to a three-phase motor?* B. R.

Answer—The opening of a fuse on either side of the transformer bank would leave the motor operating single-phase. If the motor is lightly loaded it will continue to run until shut down, but cannot be started again until the fuse is replaced. While the motor continues to operate single phase it will take about 100 per cent. more current in the active phase than when operating three-phase. The single-phase pull-out torque will be from 30 to 40 per cent. of what it was for three-phase operation.

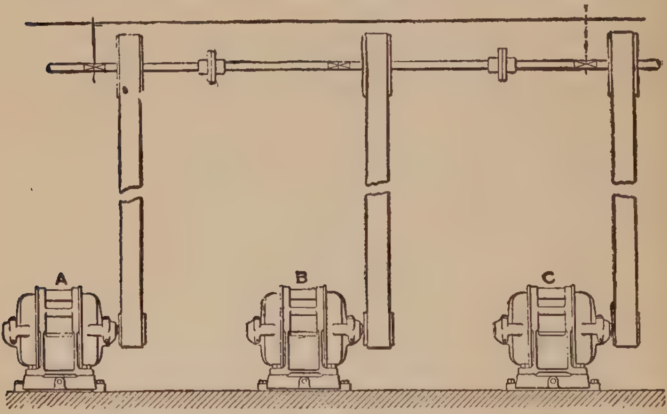
QUESTION [**Electrical Input to Develop 15 Brake Horsepower**]—*Will an electric-driven pump that requires* 15 *h.p. when driven by a* 20-*h.p. motor consume* 20 *h.p. or will it only consume* 15 *hp.?* C. D.

Answer—The actual, or brake, horsepower developed by the motor will be the number of horsepower required by the pump, regardless of the rated capacity of the motor. The amount of electric power absorbed by the motor would depend on the efficiency of the motor when developing 15

brake horsepower. A 20-hp. motor developing 15 brake horsepower would have an efficiency of about 80 per cent, requiring an electrical input of $15 \div 0.8 = 18.75$ electric horsepower, or $18.75 \times 746 = 13{,}987.5$ watts.

QUESTION [Unbalanced Load on Induction Motors] *We have in our plant three 75-hp. induction motors driving a lineshaft, as shown in the figure. When motor A is carrying the entire friction load, the ammeter reads 47 amperes per terminal; when A and B motors are running, the ammeter on the former shows 42 amperes per phase, and when all three motors are driving the friction load, the ammeter on motor A shows 35 amperes. When all three motors are driving the full load on the lineshaft, motor A takes 135 amperes, motor B 85 amperes and motor C 75 amperes. The motors are operating on a two-phase 30-cycle 425-volt system. I would like to know why motor A takes more load than either of the other two machines and what may be done to balance the load between the different motors. Motor A operates at a higher temperature than motors B or C.* G. A. B.

Answer—Since motor A is drawing more current and operating at a higher temperature than the others, it is apparent that either the characteristics or the full load slip



GROUP OF MOTORS BELTED TO LINESHAFT

of the three motors are not the same and that motor A is taking more than its share of the load. When motor A is driving the friction load alone, it takes 47 amperes. The small reduction in this current as the other motors are added is due to two causes. As the mechanical load is reduced on an induction motor, its power factor is also reduced, consequently the current will not decrease in proportion to the load reduction; and the other cause for the small reduction of current in motor A is that this motor is taking more than its share of the load.

An arrangement where two or more motors are used to drive the same load cannot be expected to divide the load equally between the different motors, since it is not possible to design and build two machines that possess identically the same characteristics. Even if the machines had the same internal characteristics, the difference in the external conditions would in all probability affect the load division. The latter will be affected mainly by the size of the pulleys, as affecting the slip; but also by the position of the rotor in the stator bore, which will be changed as the bearing wears; the voltage at the motors' terminals will change slightly with the location of the motors; the tension on the belt, if the machine is belt-driven; again, the resistance of the rotor may change by some of the connections between the end rings and bars becoming defective; if the motors are wound-rotor machines, the load characteristic of the machine will be affected considerably by the resistance of the brushes on the slip rings, the machine with the highest-resistance brushes taking the least load.

If the machines are of the wound-rotor type, it may be possible to equalize the load between them by putting new brushes on motors B and C, having a lower resistance than the ones now in use, or putting higher-resistance brushes on motor A. In any case the load can be balanced up by slightly increasing the size of the pulleys on motors B and C or decreasing the size of the pulley on motor A, especially if the latter is made of wood or fiber. However, until there is no danger of one of the motors becoming overloaded, there is not any serious reason why the load should be balanced on the machines. On the other hand, if the load is no gretaer at any time than that given in the problem, there is no reason why, if the load is properly balanced, two of the motors cannot carry it and they will then operate at a somewhat improved power factor over that obtained with the three machines.

All current readings should be made with the same instruments, or if different instruments are used on each motor, they should be checked "before and after" taking readings, to make sure that all instruments read alike.

QUESTION [Determining Size of Motor Required for Fan Blower]—*We have a 42-in. diameter fan blower which, when driven at 750 r.p.m., will handle 20 to 25 tons of ensilage per hour and elevate it 30 ft. According to the manufacturers, the blower doing that work required four to six horsepower. We have increased the speed to 1200 r.p.m. and handle the same quantity of material per hour elevated 62 ft. The blower is driven by an engine and we wish to change to motor drive. What size of motor should be provided?* B. H.

Answer—The data and conditions are not sufficiently complete and exact for deciding the probable characteristics of operation of the blower. The best method of determining the size of motor to be used would be to ascertain the power

required by indicating the engine while carrying a constant load with and without the blower. The difference will be the power required for the blower. Another method is to measure the power with some form of transmitting dynamometer. Other ways of testing would be by substitution of an ascertainable brake load. If the power taken by the blower is more than about one-fourth the capacity of the engine, run the blower doing regular work, using the regular throttle valve for starting and stopping the engine, and with a stop-valve between the regular engine throttle and the boiler, but with the regular throttle wide open, throttle down the supply of steam to a point where the blower doing regular work will be driven only just up to proper speed. Then, without changing the position of the stop-valve, stop the engine by closing the regular throttle valve, remove the belt from the blower and apply a prony brake to a pulley on the blower countershaft and determine the greatest brake power that can be developed by the engine with full throttle, the same opening of the stop valve and the same steam pressure as before on the boiler side of the stop-valve.

If it is impracticable to thus employ a stop valve for gaging the flow of steam under identical conditions, so as to obtain the same development of power when the engine carries the substituted brake load, another method is to drive the blower countershaft with a slack belt brought to the necessary tension with a weighted tightener-idler pulley by means of which the tension of the countershaft driving belt may be made only just sufficient for driving the blower to the proper speed. It then can be ascertained how much prony brake load can be laid on the countershaft as a substitute for the blower load under the same conditions of belt tension.

QUESTION [Transformers Connected Delta and Star]—*Will three transformers connected delta have the same capacity if connected star?* M. A. B.

Answer—When transformers are connected in delta as in Fig. 1, the current per terminal is 1.732 times the current per

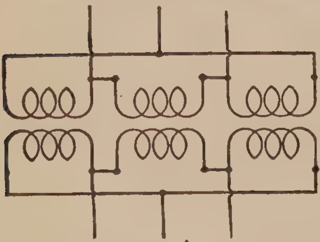


FIG. 1

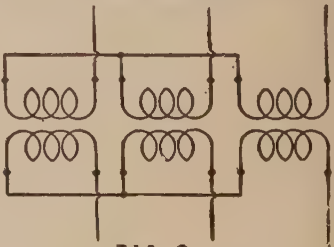


FIG. 2

transformer and the voltage between terminals is that across each transformer, that is, if the current in each transformer winding is 100 amperes, the current per terminal will be

$100 \times 1.732 = 173.2$, and if the volts across each winding is 1000, the pressure between terminals is also 1000. In the star connection, Fig. 2, the current per terminal is that per winding, but the voltage between terminals equals the volts per transformer times 1.732. In other words, if the current per winding is 100, the amperes per terminal will also be 100 against 173.2 for the delta connection. With 1000 volts across each transformer in the star connection, the volts between terminals will be $1000 \times 1.732 = 1732$, compared with 1000 for the delta connection. The total volt-amperes in each case equals volts $\times$ amperes $\times$ 1.732, which gives for the delta connection $1000 \times 173.2 \times 1.732 = 300,000$, and for the star connection $1732 \times 100 \times 1.732 = 300,000$, giving the same capacity in each case. However, it is evident that for this to be possible the volts applied to the star-connected bank of transformers must be 1.732 times as great as for the delta connected. A bank of three transformers connected delta and operating on a 6600-volt circuit will have the same capacity as the same bank connected star and operating on 11,000 volts, that is, $11,000 = 6600 \times 1.732$, approximately.

Transformers

QUESTION [Shell and Core Type Transformers]—*What is the difference in construction in shell-type and core-type transformers?* K. B. A.

Answer—In the shell-type transformer the coils are assembled and then the core is built up, of thin sheets of soft iron, around the coils, as shown in Fig. 1; and in the core-type the coils are placed around a core built up of thin sheets of soft

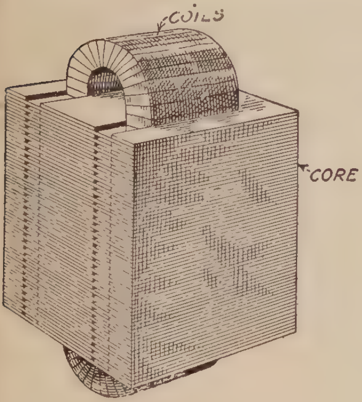


FIG. 1

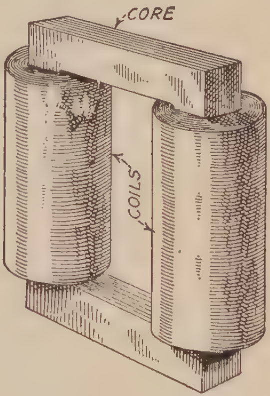


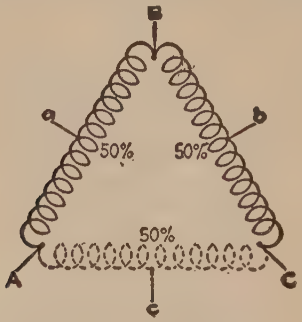
FIG. 2

iron in the form of a square link, as shown in Fig. 2. The arrangement shown in Fig. 2 is the one generally used for core-type transformers, and the one given in Fig. 1 is only one

of a number of arrangements for the core and coils of shell-type transformers, but they all follow the general idea set forth in this figure.

QUESTION [Voltage Taps on Three-phase Transformers]—*Why is one-half line voltage obtained between 50 per cent taps on a delta-connected three-phase transformer, when there are the same number of turns between the 50 per cent taps as between the full voltage terminals?* M. P

Answer—Consider one leg of the transformer, say *AC*, removed. We now have an open-delta connection. The volt-



age *AC* is the resultant of voltages *AB* and *BC*, yet it is equal to each of these and not to their sum, because they are 120 deg. out of phase. The voltage between 50 per cent taps is the resultant of half of the voltages *AB* and *BC* and is therefore equal to one-half of *AC*, or half of the line voltage.

QUESTION [Primary and Secondary of Transformers]—*Is the high-voltage winding of a potential transformer always considered the primary? If a transformer is designed to step 2300-volt alternating current down to 115, and it is connected to step 115-volt current up to 2300 volts, which winding will be the primary?* M. A. B.

Answer—The winding of a transformer which is connected to the source of power is the primary, and the winding connected to the load is the secondary. When the transformer referred to in the question is connected to step from 2300 down to 115 volts, the 2300-volt winding is the primary. When it is connected to step from 115 up to 2300 volts, the 115-volt winding is the primary and the 2300-volt winding the secondary.

QUESTION [Twenty-five-Cycle Transformers on 60 Cycles]—*Will 25-cycle transformers 6,600-volt primary 220-volt secondary work on 60 cycles 6,600-volt primary 220-volt secondary?* E. R. G.

Answer—Twenty-five cycle transformers can be operated on 60 cycles, but 60-cycle equipment cannot be operated on 25

cycles. For the same output the 25-cycle equipment will be considerably larger than the 60-cycle, consequently where 25-cycle transformers are operated on a 60-cycle circuit, the unit would be considerably larger than a 60-cycle transformer of the same capacity. When the 25-cycle transformers are used on 60 cycles, the exciting current and the flux density in the coil will be only about four-tenths what it is for the 25-cycle circuit. When 60-cycle equipment is operated on a 25-cycle circuit, the exciting current and flux density will theoretically be 2.4 times that of the 60-cycle circuit. However, due to the saturation of the iron core, the exciting current may be many times this value, and will cause the transformer to overheat even when operating without any load.

Rotary Converters

QUESTION [**A. C. Voltage on Rotary Converter**]—*If the direct-current voltage of a six-phase rotary converter is 250 volts, what must the alternating-current voltage be at the collector rings?* C. A. R.

Answer—The value of the voltage at the collector rings of a six-phase rotary converter to give 250 volts at the commutator will depend upon how the converter is connected to the transformers. If in double delta, the voltage at the collector rings will be the direct-current voltage times 0.612, or in this case $250 \times 0.612 = 153$ volts. If a diametrical connection is used, then the alternating-current volts equal the direct-current volts times 0.707 or $250 \times 0.707 = 176$. A third arrangement is the six-phase connection in which the alternating-current volts is equal to the direct-current volts times 0.354; in this problem $250 \times 0.354 = 88$ volts. These voltages are theoretical values; in practice the voltages will probably be about 5 to 10 volts higher. The connection generally used is the diametrical arrangement as this allows employing the highest alternating-current voltage at the collector rings.

QUESTION [**Inverted Converter**]—*Can a rotary converter be used to tie an alternating-current system and a direct-current system together so that power may be transferred from one system to the other?* C. M. B.

Answer—If the converter is shunt wound, it may be used for this purpose, but means must be provided to prevent the machine from racing in case it is supplying power to an alternating-current load of low lagging power factor. A wattless lagging current supplied from the collector rings of the converter will weaken the field pole and cause the machine to race unless it is operating in parallel with other alternating-current machines. Converters operated in this way usually have their field coils excited from a direct-current generator driven directly from the converter's armature shaft. Then, when the converter attempts to race, the voltage of the exciter is increased and that, in turn, increases the field

current and neutralizes the demagnetizing effect of the wattless current supplied by the armature.

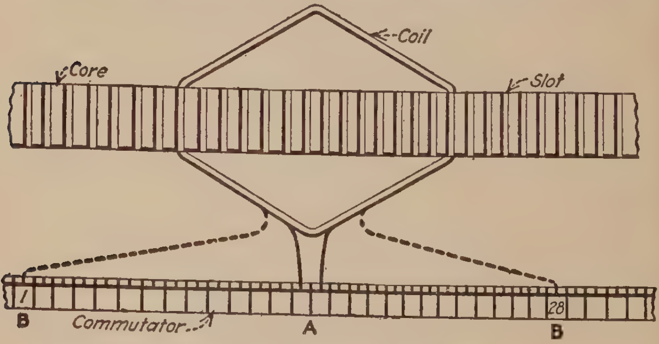
QUESTION [Rotary Converters in Parallel]—*In our plant we have two rotary converters; one machine is a six-phase and the other a three-phase unit, operating at 600 volts on the direct current side. Can these two machines be connected in parallel?* C. T.

Answer—There is no reason why these machines cannot be connected in parallel. Of course the connection will depend somewhat upon their type and method of voltage regulation. It will also be necessary to provide protection to prevent one machine from motorizing the other. This is generally accomplished by equipping the circuit breakers so they can be tripped by reverse-current relays.

Windings: Direct-Current, Alternator, Synchronous Motors and Induction Motors

QUESTION [Reconnecting Direct-Current Generator]—*We have a 120-volt direct-current generator and wish to reconnect it to operate on 220 volts. The machine has four poles, and in the armature there are 56 coils, and 56 segments in the commutator with the coils grouped in a four-circuit winding. Can the fields and armature coils be regrouped to give the higher voltages?* W. O. R.

Answer—By regrouping the armature coils into a two-circuit, or series winding, it will develop double the present voltage with the same field density, or 240 volts. The difference between 240 and 220 or 20 volts can be easily taken care of by adjusting the field rheostat. As the coils are now



grouped, the leads of each coil are connected into adjacent segments, as shown at *A*, in the figure. Fifty-six coils cannot be connected into a series winding, but 55 coils can, therefore it will be necessary to leave one coil dead in the winding and connect two of the segments together. The leads of the coils will then be connected so as to span 27 segments, or from 1 to 28, as at *B* in the figure. In order that the filed coils will

operate on the new voltage, they will have to be rewound with the present weight of wire of one-half the cross-section of that now used in the coils. In all probability a new field rheostat will be required for the higher voltage, to obtain proper adjustment of the voltage.

QUESTION [Reconnection of Armature Winding]—*What will be the correct pitch of the coil leads on the commutator of a single-series-wound armature having* 180 *coils in the winding and* 180 *segments in the commutator? The machine has* 6 *poles and operates at* 550 *r.p.m. on a* 250*-volt circuit. At present the armature is wound single-series and is giving considerable trouble by heating excessively and also sparking at the commutator. Is there any way that I can tell if the armature should be wound single, double- or triple-series?* C. R. S.

Answer—If the armature has 180 coils it cannot be connected single-series since the number of coils must agree with the formula,

$$C = \frac{P \times X}{2} \pm 1$$

where C equals number of coils in the winding, P the number of poles and X an integer which equals the pitch of the leads on the commutator and is obtained by dividing the commutator segments by the pairs of poles, in this case, $180 \div 3 = 60$; that is segments 1 and 61. Then the number of coils $C = \frac{6}{2} \times 60 \pm 1 = 181$ or 179. There are 180 coils in the winding, and since only 179 or 181 can be connected single-series, there is only one thing to do; namely leave one coil dead and use 179. This will also require the bridging together of two segments in the commutator, thus obtaining 179 segments.

On account of the speed and voltage of the machine, it is very probable that a single-series connection is correct for the armature. If a single-series connection is correct, and the machine was connected double-series, the speed would be double that of normal, and with a triple-series connection the speed would be three times normal. However, a triple-series winding would not be used on a 6-pole machine, since the same results can be obtained with a single-parallel winding and the latter would be free from all the compilcations of the former.

QUESTION [Changing Voltage of Alternators]—*We have a* 6,600*-volt* 3*-phase* 60*-cycle alternating current generator operating at* 3,600 *r.p.m., with a* 36*-coil winding, connected series star. What voltage can be obtained by connecting the winding two-paralleled star, series delta and two-parallel delta? We have also a second machine that runs* 3,600 *r.p.m. and generate* 3*-phase,* 60*-cycle* 600*-volt current. The winding in this machine has* 48 *coils connected two-parallel delta. What voltage will be generated when the winding is connected series star, two-parallel star and series delta?* W. W.

Answer—Since the first machine generates 6,600 volts when connected series star it will develop one-half of 6,600 when connected two-parallel star or 6,600 ÷ 2 = 3,300 volts. When connected series delta the voltage will be that of the series star connection divided by 1.732, in this case 6,600 ÷ 1.732 = 3,810 volts. With a two-parallel delta connection the volts will be one-half that obtained with a series-delta connection or, 3,810 ÷ 2 = 1,905 volts.

If the second machine generates 600 volts when connected two-parallel delta, then it will generate 600 × 2 = 1,200 volts when connected series delta. With a series-star connection it will generate a voltage equal to that obtained from a series-delta connection times 1.732, in this problem 1,200 × 1.732= 2,078 volts. The two-parallel star connection will give one-half that obtained from the series-star grouping or, 2,078 ÷ 2 = 1,039 volts. It may be possible to so change the speed, excitation and the winding pitch of these machines as to obtain a standard 2,200 or 2,300 volts, but before trying this it would be advisable to consult the manufacturers.

QUESTION [**Reconnecting Induction Motor**]—*We have a 60-cycle 440-volt 150-hp. 720 r.p.m. motor. This machine when it came from the factory was connected five-parallel for two-phase. Without removing any of the coils somebody connected it five-parallel-star for three-phase. In the two-phase winding the coils were arranged in groups of four. In the three-phase winding one coil has been left dead in each group, and this motor is operating satisfactorily. In other words, there are now only three active coils in each pole-phase group. What effect has the change on the horsepower rating and speed of the motor?* W. H. K.

Answer—A two-phase motor connected star for three-phase operation, using the same grouping of coils as for the two-phase machine, will be correct to operate on 125 per cent of the two-phase machine's volts. With the motor in question, the winding was grouped in five parellels for two-phase and was correct to operate on 440 volts. If this same winding was grouped five-parallel-star for three-phase it would be good for 125 per cent of 440 volts or 550. However, 25 per cent of the coils have been left inactive in the winding, consequently the motor is now only good for operation on 75 per cent of 550 or 412.5 volts, and will act the same as though it was on a circuit of 106 per cent normal volts. No doubt the motor is conservatively enough designed so that it will operate satisfactorily under the new condition. Since the copper has been reduced by 25 per cent., the output will be normally reduced by 25 per cent and the rating will be 75 per cent of 150 or 112.5 horsepower. Since the motor is operating satisfactorily under conditions equivalent to 6 per cent increased voltage, it is probably good for about 80 per

cent of its normal rating, or 120 horsepower. The starting torque and pull-out torque will be only about 80 per cent of that developed by the two-phase winding, but the speed will be practically the same for either two- or three-phase. To materially affect the speed the frequency or the number of poles would have to be changed.

QUESTION [Reconnecting Synchronous Motor]—*We have a* 6,600-*volt, three-phase,* 60-*cycle* 720-*r.p.m. synchronous motor connected one circuit "Y." The machine has* 60 *slots in the stator; two slots per pole per phase. We would like to reconnect this motor so as to operate on a* 2,200-*volt circuit at* 720 *r.p.m. How should the windings be connected?* W. H. K.

Answer—The machine as it is now connected has 10 poles; 2,200 volts, is one-third of 6,600, therefore the winding should be reconnected, for 2,200 volts, so as to give one-third the number of turns in series in the 6,600-volt winding, or 3-parallel star. However, the connection cannot be obtained since the machine has 10 poles, and 10 is not divisible by 3. The only other connection that can be made which offers a

TEN-POLE TWO-PARALLEL DELTA CONNECTION

possibility of satisfactory operation is a 2-parallel delta, as in the figure; but this is good for only $\frac{6,600}{2 \times 1.732} = 1,905$ volts. With the winding connected 2-parallel delta and operating 2,200 volts, it will be working at $\frac{2,200 \times 100}{1905} - 100 = 15$ per cent. overvoltage. This overvoltage would in general be excessive, therefore it is advisable to take up with the manufacturer the question as to the advisability of subjecting the motor to this overvoltage. The full pitch of the winding would be 6 slots; that is, the coils would be wound in slots 1 and 7. If the coils happened to be wound in slots 1 and 6, the cord factor would be 0.95. Increasing the pitch of the winding to 0 to 7, or full pitch, would make it suitable for operation on $\frac{1905}{0.95} = 2,005$ volts, or the motor would then be operating at 10.7 per cent. overvoltage. If this condition can be met, there is little doubt that the motor will operate on 2,200 volts.

QUESTION [2200-Volt Motor on 440 Volts]—*What horsepower would a three-phase 50-hp. 2200-volt 750-r.p.m. motor develop if connected on a 440-volt line?* P.F.M.

Answer—Theoretically, one-fifth of 50, or 10, hp. and the starting torque would be 1/25 of that at 2200 volts. The only thing to do is to reconnect the stator winding for 440 volts. Just what the grouping should be cannot be determined without knowing the number of coils and how they are grouped in the 2200-volt winding. If the 2200-volt winding is connected in series star, which is probably the case, then the 440-volt winding may be connected 5-parallel star. However, if the number of coils per phase cannot be divided by 5 it may be somewhat of a problem to distribute the coils so as to get a satisfactory winding. A 3-parallel delta connection might be found to give a more satisfactory grouping. This gives an equivalent 418-volt winding, which is close enough to 440 volts to operate satisfactorily.

QUESTION [Induction-Motor Winding Bracing]—*Why are insulated rings placed over the ends of induction motor coils where they extend beyond the stator core and why in some cases are they placed on both ends and in others one one end only?* K. A. R.

Answer—Frequently, where the coil throw is long or if the coils are not mechanically rigid, the ends are liable to be distorted when a heavy current flos through them. To prevent this, insulated rings are placed around the outside of the winding and the coil ends laced to these rings with heavy cord. This practice has been found to increase the life of the coils, since it tends to prevent any vibration of the coil ends that would otherwise occur. Where the throw of the coils is short the rings are only put on the end of the winding opposite to

the one where the cross-connections are made, the cross-connection giving the necessary support to the coil ends on the end of the stator winding where the connections are made.

QUESTION [Chord Factor]—*What is the chord factor of a coil in an induction-motor winding and how is it figured?*

C. A.

Answer—The chord factor of any coil in an electrical machine generating an electromotive force is a factor representing the effectiveness of the coil in generating voltage. The chord factor is equal to the sine of one-half of the angle in electrical degrees which the coil spans. In a stator core having 72 slots wound for 6 poles, for the coils to span full pitch they would have to span $72 \div 6 = 12$ slots, that is, the sides of a coil would have to lie in slots 1 and 13. This would represent 180 electrical degrees and the sine of one-half of 180 degrees is 1, hence, the chord factor of the coil is 1, and the coil is 100 per cent. effective in generating a counter-electromotive force. Each slot equals $180 \div 12 = 15$ electrical degrees. Then if the coil lay in slots 1 and 11 it would span only 10 slots or $15 \times 10 = 150$ electrical degrees, and the chord factor would be the sine of one-half of 150 deg., or 75 deg., equals 0.96. This means that the winding is only 96 per cent. as effective in generating a counter-electromotive force in the latter case as it was in the former. If with full pitch, under a given condition, the winding was good for operation on 500 volts, then with the coils in slots 1 and 11 the winding would be good for only $500 \times 0.96 = 480$ volts.

QUESTION [Induction-Motor Rotor Connections]—*I have rewound the rotors of a number of wound-rotor-type induction motors and in each case the winding was connected star. Can the delta connection be used in grouping these windings? If so, why is it not used more generally?* M. T. R.

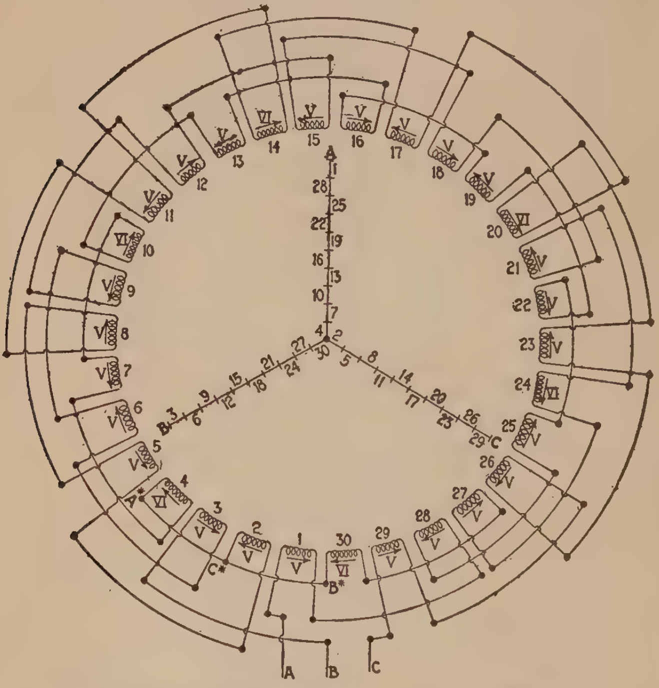
Answer—Either connection may be used, but in small and medium motors the star connection gives a more satisfactory winding design. The star connection gives a terminal voltage 73 per cent higher at the slip rings than the delta connection. This is very desirable where the rotor is bar wound, since with this type of winding the voltage is comparatively low, and the current relatively high. With the star connection the current will be 58 per cent of that obtained with the delta connection. Consequently, the starting resistance is much cheaper to construct. In some large-sized motors, however, the voltage produced in the rotor winding with a star connection may be so high as to be difficult to insulate and also require specially designed central equipment to make them safe for an operator to handle. In such cases the rotor is connected delta, as this grouping generally gives a voltage at the slip rings that is of a safe value.

QUESTION [Uneven Number of Coils in Rotro Phases]—*In a 40-hp. three-phase wound-rotor motor in which*

I short-circuited the slip ring and tried starting with a 50-hp. compensator, the motor developed positions where it would not start; in fact, the operation of the motor was so unsatisfactory that operating it under the foregoing conditions was abandoned. The stator has 120 coils grouped in a 10-pole winding, and the rotor has 156 coils connected two phases of 57 coils each and one phase of 42 coils, connected series star. Why is it that there is not 156 ÷ 3 = 52 coils in each phase? Has the unbalancing of the winding anything to do with the trouble that the motor developed? M. P.

Answer—It is apparent that a mistake has been made in connecting up the windings; two pole-phase groups of 5 coils each being taken out of the phase that has only 42 coils and connected into the two phases that have 57 coils each.

With 156 coils, in a 10-pole rotor, to give a balanced winding there will be 52 coils per phase, grouped 8 poles of 5 coils each, and 2 poles of 6 coils each; the 6-coil pole-phase groups will have to be distributed as in the figure. The Roman notations indicate the number of coils in each pole-phase group.



THREE-PHASE SERIES-STAR GROUPING

There is little doubt that the way the rotor was connected

contributed to the unsatisfactory operation of the motor, especially at starting. However, after the motor was started it should have given satisfactory service unless there was some trouble in the stator winding or connecting wires.

Miscellaneous

QUESTION [Motors and Lights on the Same Circuit] —*What are the objections to supplying electric motors and lamps from the same feeder provided they are of the same voltage?*

J. N.

Answer—The chief objection to this practice is that if the motors are started under heavy load, they will cause the voltage to drop to such an extent as to make the lamps fluctuate. The most satisfactory voltage for motor operation is 240 or higher, while lamps operate most efficiently around 120 volts. Consequently, if the motors are to be designed and operated at the higher voltage, the lamps must be operated in series groups, which to say the least is not satisfactory.

QUESTION [Raising the Capacity and Power Factor of a System]—*We are planning to add a 1,500-kva. turbo-unit to our 3,000-kva. generating equipment, and also wish to install a synchronous condenser to improve our power factor, now only about 70 per cent. What should be the capacity of the synchronous condenser and the capacity of an exciter to supply both the new generating unit and the condenser?* M. H.

Answer—Assuming that the condenser will not be used for power, but will operate under no load, and 100 per cent. power factor is desired; also that the power factor of the future load will remain 70 per cent:

The wattles kva. drawn by the load which the condenser must supply will be: 4,500 kva. $\times \sqrt{1 - 0.70^2} = 3{,}200$ kva. With 5 per cent. losses in the machine, your condenser capacity should be $\frac{3{,}200}{0.95} = 3{,}400$ kva. However, it is poor practice to attempt to bring the power factor above 90 per cent. when installing apparatus for this specific purpose, as the advantages gained in operating characteristics would not warrant the greater cost of a larger machine. The necessary capacity of a condenser to raise your power factor to 90 per cent would be,

$$\frac{4{,}500}{0.95}\left(\sqrt{1 - 0.70^2} - \sqrt{1 - 0.90^2}\right) = 1{,}320 \text{ kva.}$$

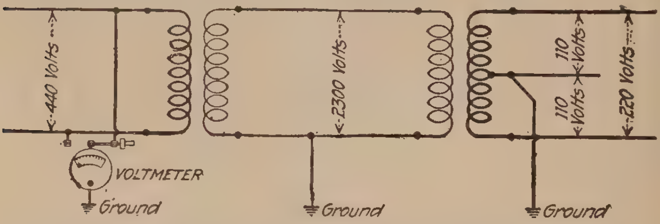
or little more than one-third of the capacity necessary to give 100 per cent power factor.

The current and voltage characteristics of the exciter suitable for any machine should be obtained from the manufacturer, but it is safe to roughly assume an exciter capacity

of 3 per cent of full load for the generator and 4 per cent of full load for the condenser.

QUESTION [Ground Detector Voltmeter]—*We have a lighting system stepping up from 440 volts through a transformer to 2300 volts for transmission and then stepping down to 110-220 volts for lighting service. If a ground occurs on either the 2300- or the 110-volt line, will it indicate on the ground-detecting voltmeter on the 440 supply?* E. K.

Answer—It is evident from the figure that a ground on either the 2300-volt or the 110-volt system will not be indicated on the ground-detecting voltmeter on the 440-volt supply, since each system is insulated from the other. How-



ever, the 2300-volt system should be kept insulated from ground, since a ground on the high voltage subjects the insulation between the primary and secondary windings to the full voltage of the transformer.

QUESTION [Choke-Coil Voltage]—*What determines the voltage rating of choke coils such as are used with lightning arresters?* W. A. M.

Answer—The voltage rating of a choke coil used with lightning arrester has in general no reference to the coil, but to the insulators on which the coils are mounted. A coil designed for 100 amperes would be practically the same whet-er used on a 11,000-volt circuit or 33,000 volts, but the insulators for the 33,000-volt coil would have to be designed to withstand three times the voltage of the 11,000-volt coil. From a protective standpoint the 33,000-volt coil would be perfectly stisfactory on a 6,600-volt circuit, but from the commercial side the cost of insulators would be necessarily high.

QUESTION [Charging Storage Battery in Lamp Circuit]—*Can a storage-battery cell be charged from a lighting circuit by cutting one wire supplying several lamps and connecting the battery in series with those lamps? If so, what will be the effect on the lamps?* W. B.

Answer—A battery may be connected in series with lamps on a direct-current lighting circuit for the purpose of charging it, provided care is taken to have correct polarity. Otherwise the battery will discharge and, if left long enough, may

be seriously damaged. If the connection gives correct polarity for charging, the lamps will be slightly dimmed as the voltage of the battery opposes that of the circuit and the voltage on the lamps will be the difference between the two. If the connection gives wrong polarity, the lamps will burn more brightly, as the voltage of the battery assists that of the line.

QUESTION [**Resistance in Series with Storage Battery**]—*A battery made up of* 16 *cells requires a charging current of* 25 *amperes. What value of resistance must be connected in series with the battery to charge it on a* 110-*volt circuit?*
B. W.

Answer—Negelecting the internal resistance of the cells, which is very low, the external resistance $R = \frac{E—e}{I}$ where E is the charging circuit voltage, e the voltage required by the battery at the beginning of charging (normally 2 volts per cell) and I the charging current. At the beginning of the charge in this problem, e will equal about $2 \times 16 = 32$ volts. Therefore the external resistance $R = \frac{110—32}{25} = 3.12$ ohms. If it is deired at any time to charge the battery at less than 25 amperes the external resistance will have to be increased in inverse proportion to the current.

QUESTION [**Electrical Conductivity of Water**]—*Is water a good or a poor conductor of electricity?* A. J. C.

Answer—Chemically pure water is an insulator of electricity, but when water is contaminated with almost any other substance it becomes a conductor.

QUESTION [**High Tension-Grounds for Icicles**]—*Could a short-circuit result from iciles forming from the eaves of a roof to one or more conductors of a* 25,000-*volt three-phase line with grounded neutrals?* E. A. T.

Answer—Icicles would not cause a short-circuit in such a case, as pure water and the ice formed from pure water are very poor conductors. Probably there would be some leakage of current from the conductor, but if this reached any appreciable figure, it would heat the icicle and melt it. Outdoor high-tension apparatus and line insulators are frequently covered with sleet and ice without disturbing the operation of the lines.

QUESTION [**Heating Water by Electricity**]—*If* 25 *kw. of three-phase electrical power, at* 220 *volts* 60 *cycles, are available at no cost, how much water may be heated per hour to temperatures ranging from* 120 *to* 200 *degrees Fahrenheit?*
L. R.

Answer—One kilowatt totally expended in heat produces 3,415 B.t.u. per hour; therefore 25 kw. will produce 25 × 3,415 = 85,375 B.t.u. per hour. If no heat were lost, this would raise the temperature of 1,000 lb. of water through 85.4 deg. F. in one hour, or 500 lb. through twice the number of degrees. It should be possible, by carefully packing the heater to reduce heat losses to about 10 per cent, and to heat about 457 lb. of water per hour from 32 to 200 deg. F., or 873 lb. from 32 to 120 deg. F.

QUESTION [Distance of Pulley Centers for Motor Drive]—*What is the approximate rule for the minimum distance between pulley centers for open belt drives from small electric motors?* H. P. R.

Answer—A rule that has given satisfaction with pulleys of usual sizes is that the distance between the pulley centers should not be less than three times the sum of the diameters of the pulleys. A distance of four or five times the sum of diameters will result in a better drive.

QUESTION [Stream Flow Required for Hydro-Electric Plant]—*What would need to be the rate of flow of a stream with* 16 *ft. working head for development of sufficient power with turbine waterwheels for driving one* 100- *and one* 75-*kw. generator?* S. K. W.

Answer—The efficiency of the generators would be about 90 per cent. and the net efficiency of turbines and power transmission would be about 70 per cent., making an over-all efficiency of about 63 per cent., and as 1 kw. is equal to development of 44,240 ft.-lb. per min., then for each kilowatt actually generated, the water would need to develop 44,240 ÷ 0.63 = 70,222 ft.-lb. per min. With 16 ft, head each cubic foot of water used per minute would develop 16 × 62.3 = 996.8 ft.-lb., so that each kilowatt output of the generators would require a flow of water at the rate of 70,222 ÷ 996.8 = 70.45 cu.ft. per min., generation of 200 kw. would require 7045 cu.ft. per min., and 75 kw. would require 5283.75 cu.ft. per min.; or for driving both generators at the same time, the flow needs to be at the rate of 12,328.75 cu.ft. of water per minute.

QUESTION [M. E. P. of Engine for Generation of Stated Kilowatt Load]—*What m.e.p. would be necessary for a* 10x10-*in. engine running* 300 *r.p.m. for operation of a* 35-*kw. direct-current generator when developing full capacity?* G. B.

Answer—The efficiency of a generator of the given capacity under full load would be about 90 per cent, and the mechanical efficiency of an engine of the stated size and speed when driving the load given would be about 88 per cent, depending on the design, construction and adjustment. As

1 kw. = 44,240 ft.-lb. per min., the assumed efficiencies would require the steam acting on the piston to develop $\frac{44{,}240 \times 35}{0.90 \times 0.88} = 1{,}955{,}050$ ft.-lb. per min. Neglecting the reduction of piston area by the piston rod, 1 lb. m.e.p. would develop

$$(10 \times 10 \times 0.7854) \times \frac{10}{12} \times 2 \times 300 = 39{,}270 \textit{ ft.-lb. per min.}$$

so that development of 1,955,050 ft.-lb. would require

$$1{,}955{,}050 \div 39{,}270 = 49.78 \textit{ lb. m.e.p.}$$

Section VI

Refrigeration

Section VI
Refrigeration

QUESTION [Data for Calculating Duty of Refrigeration Plant]—*What data must be obtained in order to calculate the theoretical refrigeration duty secured in a refrigeration plant?* J. G. S.

Answer—Take an indicator card from the compressor or two or three of them, during any given time, say over a half-hour period, and obtain the following information: (1) Speed of the engine per minute; (2) length of test; (3) pressure of the suction elbow; (4) temperature in degrees Fahrenheit at suction elbow; (5) discharge pressure at the discharge elbow; (6) temperature of discharge vapor at the discharge elbow; (7) temperature of the liquid in the receiver; (8) if possible, temperature at the outlet of the coal.

If two suction pressures are carried and one of these suction pressures is connected to the head end and the other to the crank end of the cylinder, it will be necessary to obtain data on both. Then a close estimate can be made of the refrigerating output. However, it must be remembered that this will be based on calculations that may be wrong, due to leakage in the cylinder and variations in the volumetric efficiency of the cylinder. The best way to find the refrigerating duty secured from a plant is to weigh the brine which is being cooled by the ammonia, for by that method the actual duty is ascertained.

QUESTION [Quantity of Cooling Water]—*How can I estimate the amount of cooling water required for the condensing of our refrigerating plant?* J. M. C.

Answer—The gallons of water required to be pumped over the condenser may be obtained by the formula

$$g = \frac{(H - q)W}{(t_1 - t_2)8.33},$$

where

g = Gallons of water per minute;
H = Total heat in one pound of ammonia entering con denser;
q = Heat of the liquid ammonia leaving condenser;
W = Pounds of ammonia circulated per minute;
t_1 = Temperature of entering water;
t_2 = Temperature of ex tiwater.

The values of H, which consists of the latent heat of vaporization of the ammonia at the condenser pressure plus the superheat, may be obtained from a table of properties of ammonia. If the heat of superheat is not shown in the tables, this may be calculated by multiplying the difference between the actual ammonia temperature and the boiling temperature at the condenser pressure by 0.63.

QUESTION [Pumping Ammonia From Coils]—*Is it possible to pump ammonia from freezing coils by creating a vacuum on the compressor while the freezer temperature is* 10 *deg. below zero?* P. S.

Answer—If the temperature of the freezer is 10 deg. below zero, the liquid ammonia in the coils cannot absorb heat from the freezer and boil unless the coil temperature is 10 degrees or more below zero. The pressure of ammonia vapor corresponding to a temperature of 10 degrees is 23.75 lb. If the compressor is started, the coil pressure will be 23.75 lb. If the expansion valve is throttled so that little ammonia enters the coils, the suction of the compressor will remove part of the ammonia vapor, reducing the pressure. But as the pressure is lowered, the boiling temperature also drops, and if ammonia liquid is present, it will evaporate at the lower pressure, giving a lower coil temperature, although the freezer may still be at 10 degrees. If only ammonia vapor is present in the coil, the ireduction of pressure will tend to superheat the vapor.

QUESTION [Fuel Saved in Refrigerating Plant from Use of Cooling Water of Lower Temperature]—*We have two double-acting vertical ammonia compressors each* 10 *in. in diameter by* 30-*in. stroke running* 65 *r.p.m., operating under a back pressure of* 20 *lb. gage in the suction line and average pressure on the compressors* 185 *lb. per sq. in. The compressors are driven by a horizontal non-condensing Corliss engine with cylinder* 20 *in. in diameter in which the m.e.p. is* 55 *lb. per sq. in. The temperature of cooling water going over the ammonia condensers is* 80 *deg. F. How is it determined what the saving of fuel would be from the use of cooling water having a temperature of* 54 *deg. F.?* W. W. H.

Answer—The best way of solving this problem is by the use of a Mollier heat chart. It is assumed that the liquid ammonia is at 96 deg. F., which is the boiling temperature at 200 lb. absolute (185 lb. gage). In this condition each pound of ammonia has in it 150 B.t.u. measured from − 40 deg. F. On evaporating at 20 lb. gage (35 lb. abs.) each pound of dry saturated ammonia vapor contains 614 B.t.u., or 464 B.t.u. have been absorbed from the coils. After compression from 35 lb. absolute to 200 lb. each pound contains 722 B.t.u., or 108 B.t.u. of work has been done for each 464 B.t.u. of refrig-

eration, or for each B.t.u. of refrigeration 0.23 of B.t.u. work is required.

If the cooling water is at 54 deg. instead of 80 deg. and the same temperature difference of 16 deg. between the water and ammonia in the condenser is assumed, the liquefying temperature of the ammonia in the condenser will be 70 deg. which corresponds to 127 lb. per sq. in. absolute.

A pound of liquid ammonia at 127 lb. contains 120 B.t.u. After going through the expansion valve and evaporating at 35 lb. absolute, it contains 614 B.t.u., or 494 B.t.u. has been picked up from the coils. After compression, back to a discharge pressure of 125 lb., each lb. of ammonia vapor contains 691 B.t.u. In other words, $691 - 614 = 77$ B.t.u. has been added by compression to do 494 B.t.u. of refrigeration. This means that each B.t.u. of refrigeration requires 0.15 B.t.u. of work. The work required per ton of refrigeration with the low-temperature water will be $\frac{15}{23}$ of that required with the high-temperature water. The amount of steam saved per hour depends upon the efficiency of the engine at the new cutoff, but under new circumstances the steam consumption should be about $\frac{15}{23}$ of that required under the high-tempera ture water.

QUESTION [Brine Circulation for Separate Refriggerating Rooms]—*For a 15-ton refrigeration plant consisting of an icemaking tank with a capacity of 5,000 lb. of can ice per 24 hours and refrigerator and cold storage rooms, to be cooled by circulation of brine, is it more desirable to have a loop system of brine piping in connection with the ice tank, or a separate supply and return main to refrigeration room?* L. B. W.

Answer—Better control will be obtained with separate supply and return circulation for the refrigerating rooms supplied from a shell and tube brine cooler separate from the ice tank.

QUESTION [Determination of Ammonia in Calcium-Chloride Brine]—*What is the best method of determining the presence of ammonia in calcium-chloride brine?* R. L. B.

Answer—Nessler's reagent will prove satisfactory. This solution is made up as follows: 17 grams of mercuric chloride is dissolved in about 300 c.c. of distilled water to which is added 35 grams potassium iodine dissolved in 200 c.c. of water, constantly stirred until a slight permanent red precipitate is produced. To the solution thus formed add 120 grams of potassium hydrate dissolved in 200 c.c. of water; allow to cool before mixing. The amount is then made up to one liter by addition of water and mercuric chloride until a permanent precipitate again forms. After standing for a short time the solution is ready to be used. When a few drops of

this solution are added to a sample of calcium-chloride brine or water in test tubes, if there is any ammonia present a slight coloration of the liquid will take place. A large quantity of ammonia will make the color still darker.

QUESTION [Rating of Refrigerating Machine]—*What is meant by a twenty-ton refrigerating machine?*
W. L. A.

Answer—A twenty-ton refrigerating machine is one that produces a refrigerating effect at a rate equal to the melting of twenty tons of ice per 24 hours. A pound of ice in melting requires a quantity of heat known as the latent heat of fusion, which amounts to 144 B.t.u., and twenty tons of ice would absorb $20 \times 2{,}000 \times 144 = 5{,}760{,}000$ B.t.u.

QUESTION [Foreign Gas in Refrigeration System]—*How can I determine if there are foreign gases in our refrigeration system?*
W. L. M.

Answer—If the capacity of the plant has dropped and the head pressure persists in keeping high, foreign gases are usually the cause. To detect such gases insert a hose connected to the condenser purge valve in a pail of water. The ammonia gas will be absorbed by the water, while any foreign gas will bubble. If inflammable the gas can be ignited with a match as it leaves the water.

QUESTION [Pipe Joints for Ammonia Systems]—*Is it more advisable to use screwed or welded joints in ammonia systems?*
W. F. L.

Answer—Welded joints are preferable, but screw joints are sufficient if the pipe is sweated into recessed fittings.

QUESTION [Removal of Water from Ammonia Refrigeration system]—*From blowing out of a gasket on an old type of ice machine, we are troubled with considerable water in the ammonia. Could not the water be removed by blowing out the trap on the brine cooler, or is a regenerator necessary for separating the water, and if so what is the correct temperature?*
N. E.

Answer—For removal of the water with economy of ammonia it is necessary to employ a regenerator or purifier. The water cannot be eliminated by blowing of the trap at the brine cooler, as ammonia is highly soluble in water, since water will absorb 400 to 600 times its own volume of ammonia vapor. After water has circulated in the system a short time, it becomes aqua ammonia, and if blown out at the trap, there is loss of a large volume of absorbed ammonia vapor.

By properly heating the liquid in a regenerator, the ammonia is driven off while the water remains, to be discharged separately. It is impossible to eliminate all the water, because in driving off the ammonia, a small amount of water vapor is also driven off. The action of the regenerator is the same as that of a generator in an absorption system. The correct temperature will depend on the pressure carried in the regenerator, which, in most installations, is the pressure on the suction side of the system.

Books on refrigeration quote a table which gives the correct temperature and corresponding steam pressure for any generator pressure required in an absorption system, and these pressures hold for the regenerator or reclaimer. If the regenerator is connected in the discharge side of the system, it will be necessary to have the temperature of the steam coils high enough to cause the ammonia to gass-off at the higher pressure

QUESTION [Use of Ammonia Fore Cooler]—*Does it pay to install a fore cooler for the liquid ammonia before it reaches the expansion valve?* G. T.

Answer—Since the ammonia in boiling in the coils mu best cooled before any refrigerating effect is secured it is advisable to install a cooler whereby the ammonia gas passing from the coils to the compressor chills the liquid ammonia.

QUESTION [Metallic Rod Packing for Old Compressor]—*Can metallic packing be used on an old ammonia compressor piston rod?* R. A.

Answer—If the rod is not scored or shouldered, there is no reason why metallic packing suitable for ammonia should not be used.

QUESTION [Frosting of Suction Line]—*How far should the suction side of a refrigerating system be frosted back?* H. S. C.

Answer—Most engineers find it best to carry the frost back to the suction-valve connection, if this does not make the discharge line too low in temperature.

QUESTION [Refrigeration from Brine Coil]—*We have a calcium chloride brine coil in an ice box operating under the following conditions:*

Average incoming temperature of brine, deg. F......	23.
Average outgoing temperature of brine, deg. F......	24.75
Range in brine temperature, deg. F................	1.75
Cu. ft. of brine passing through coils in 24 hours....	857.
Specific gravity of brine.........................	1.16
Specific heat of brine............................	0.83

QUESTION—*What amount of refrigeration is produced?*

W. P.

Answer—The heat absorbed per 24 hours would be given by the formula,

$$(1)\ H = W(T_2 - T_1)c,$$

in which

H = B.t.u. absorbed per 24 hours;
W = Weight of brine passing through the coils in 24 hours;
T_2 = Final temperature of the brine;
T_1 = Initial temperature of the brine;
c = Specific heat of the brine.

Since the specific gravity of the brine is 1.16, taking the weight of 1 cu. ft. of water as 62.3 lb., would make the weight per cubic foot equal to $1.16 \times 62.3 = 72.268$ lb., and havi ng 857 cu. ft. passing through the coils in 24 hours, the value of W in the formula would be $857 \times 72.268 = 61,934$ lb. As the brine has a final temperature of 24.75 deg. F. and an initial temperature of 23 dg. F., the term $(T_2 - T_1)$ would be equal to $24.75 - 23 = 1.75$ deg. F., and as c is 0.83, by substitution,e the formula becomes

$$H = 61,934 \times 1.75 \times 0.83 = 89,959 \text{ B.t.u. per 24 hr.}$$

The unit of refrigeration is the number of B.t.u. that must be abstracted from one ton of water at 32 deg. F. to produce one ton (2,000 lb.) of ice of the same temperature. Sinc test latent heat of ice is 144 B.t.u.—that is, since 144 B.t.u. mu be withdrawn from one pound of water at 32 deg. F. to pr- h duce one pound of ice at 32 deg. F.—the unit of refrigeration called one ton of refrigeration is equal to the removal of $2,000 \times 144 = 288,000$ B.t.u. Hence for removal of 89,959 B.t.u. in 24 hours the refrigeration produced in that time is $89,959 \div 288,000 = 0.31$ ton.

QUESTION [An Agitator Pipe Freezer]—*In our rawe water ice plant the pipes that carry the agitating air into the ic cans often freeze up. What is the cause and remedy?*

D. F. C.

Answer—The amount of water vapor contained in a given volume depends on the temperature. The amount of the vapor held in suspense at 60 deg. F. is much greater than at 20 deg. F. If the air is cooled, it cannot carry as much vapor as before and some of the vapor will condense. An example of this is the familiar dew which settles on the grass on a cool night following a warm day.

In the raw-water plant, if the air passing along the agitator pipe is warm, a large amount of water vapor is carried along with the air. At the outlet in the can when the water is below freezing, a greater part of the water vapor condenses and, settling in the air pipe, freezes.

The remedy is to cool the agitating air by means of a dehumidifier before allowing it to enter the ice can.

QUESTION [Detecting Ammonia in Brine]—*The writer is of the opinion that one of the brine tanks has a slight ammonia leak. Is there a way to tell or to determine the exact amount of ammonia that this brine could contain per cubic feet? A 2-oz. sample taken from this tank will turn pink when adding 3 drops of phenolphthalein test solution.* E. S.

Answer—Phenolphthalein cannot be depended upon as indicating the presence of ammonia although it turns pink. The action is due to the alkalinity of ammonia, and it is possible that the brine itself may be slightly alkaline.

For this reason, in testing, the phenolphthalein drops should be replaced by a phenolphthalein paper, or litmus paper. Instead of immersing the paper in the suspected brine, a drinking glass should be half filled with the brine and a small amount of caustic soda added. After allowing sufficient time for the caustic soda to dissolve, the glass should be covered by a strip of glass to the underside of which is attached a piece of litmus paper moistened at the ends to cause it to adhere.

If the brine contains ammonia the paper will be turned pink by the action of the ammonia fumes. Warming the brine will hasten the action. If the paper is not affected after four or five minutes, the brine contains no ammonia. The amount of ammonia present can be approximated by determining the time elapsing before the paper changes color. As a guide a known amount of ammonia may be added to a given weight of pure brine and the time required noted.

QUESTION [Quantity of Ammonia in Refrigerating System and Time for Blowing Off Oil Trap]—*How can an idea be obtained of the amount of ammonia there is in an ammonia refrigeration system, and whether there is enough or too much in case there is no gage glass on the receiver? Also, when is the best time to blow off the oil trap, after a period of shutdown or when running?* C. F.

Answer—As long as the suction line is frosted up to the machine, one may be confident that enough liquid is reaching the evaporating coils. This is the part where the liquid is needed, and as long as this condition prevails there is ample liquid. Of course the receiver should be partly filled, to act as a reserve supply in case the refrigerating duty must be unexpectedly increased. When the oil trap is not completely blown down, it is best to blow it off when the plant is shut down.

QUESTION [Purging Ammonia System]—*For purging our ammonia system of air, we have a hose connected to the top of an oil trap and into a barrel containing about 30 gal. of water. When purging, the water rumbles as when steam is discharged*

from a hose into a barrel of water. There is little noise or odor until a fog appears, and then there is a cracking noise. We just crack the purge valve. Is this method correct? W. F. C.

Answer—Usually, the top of the oil trap is not the proper place for the purge connection. Purging should be done at the condenser. The air will cause the water to circulate as you describe. The crackling sound is caused by the sudden absorption of the ammonia vapor by the water. It is preferable to purge at the top of the condenser after shutting down and cooling the condenser as much as possible. This will condense most of the ammonia vapor in the condenser, and the ammonia lost with the purged air is less than when the plant is purged while running.

QUESTION [Preparing Sulphur Sticks and Candles] —*How are sulphur sticks and sulphur candles prepared for locating leaks in a refrigerating system?* J. L. S.

Answer—Melt the sulphur by placing it in a flat-bottomed can or pan heated to about 250 deg. F. Sufficient heat usually may be obtained by placing the vessel in direct contact with a hot steam pipe or engine steam-chest cover, or in a boiler uptake. If the melting is done over a fire or flame, the vessel should be supported on an iron plate to prevent overheating and ignition of the sulphur. For making sulphur sticks, twist together three or four strands of candle wick. Cut it up into lengths about 12 in. long, dip these into the molten sulphur for a few minutes and withdraw to cool in form of a rough candle. If candle wick is not at hand, very good sulphur sticks can be made by dipping splints of soft wood into the molten sulphur.

Better forms of candles are made in molds. A candle mold is readily improvised by rolling sheet asbestos into tubes that have their lower ends packed in fireclay and with the lamp wick stretched through the tubes. The tubes are filled with molten sulphur. When the sulphur has cooled, the candles are in convenient form for use with or without the asbestos covering.

QUESTION [Cooling Water by Pumping Through Submerged Coil]—*We are intending to submerge a 2-in. pipe coil* 1,440 *ft. long in a reservoir that holds* 500,000 *gals. of water. Water at* 120 *deg. F. is to be pumped through this coil at the rate of* 70 *gal. per min. The temperature of the water in the reservoir is* 56 *deg. F., and* 1,300 *gal. of water passes through the reservoir per minute. To what temperature may we expect to cool the water passed through the coil?* D. M. W.

Answer—The rare of cooling will depend largely on the character of circulation of the reservoir water over the exterior

of the 2-in. pipe coil. For instance, if the coil is submerged in the swiftest current of water passing through the reservoir, the cooling will be considerably faster and consequently a lower temperature will be obtained than if the coil were submerged in a part of the reservoir where the water is nearly stagnant.

Water pumped through a continuous or single 2-in. pipe coil at the rate of 70 gal. (583 lb.) per min. would have a velocity of approximately 458 ft. per min. and would pass through 1,440 ft. of pipe in 3.14 min. For a velocity of 50 ft. per min. of the reservoir water surrounding the coil, there would be transfer of about 10 B.t.u. per hour, or 0.166 B.t.u. per minute per square foot of surface per degree mean difference of temperature.

Calling the final temperature t, the mean temperature would be $(120 + t) \div 2$, and the mean temperature difference would be $[(120 + t) \div 2] - 56$. The cooling surface of 1,440 lin. ft. of 2 in. pipe would be $1{,}440 \div 1.61 =$ about 900 sq. ft., and the B.t.u. transferred in 3.14 min. required for the water to traverse the full length of the coil would be

$$3.14 \times 0.166 \times 900 \left(\frac{120 + t}{2}\right) - 56 = (120 - t)\ 583$$ for which the final temperature $t = 83$ deg. F.

QUESTION [Refrigeration Capacity for Cooling Air] —*How much refrigeration capacity is required to cool* 1,000 *cu. ft. of air per min. from* 80 *to* 40 *deg. F., and how many square feet of pipe surface are required for the cooling coils?* H. F.

Answer—In 1,000 cu. ft. of air at 80 deg. F., handled per minute, the weight of dry air would be $1{,}000 \div 13.6 = 73.5$ lb., and assuming the specific heat at constant pressure to be 0.24, cooling the air from 80 to 40 deg. F. would require the removal of $73.5 \times 0.24 \times (80 - 40) = 705.6$ B.t.u.

In addition, each cu. ft. of air contains some water vapor. If the humidity is 100 per cent, each cubic foot would contain as much vapor as contained in a cubic foot of dry saturated steam at 80 deg. F. and at the pressure of 1.029 in. of mercury, at which temperature and pressure the density is 0.00157 lb. per cu. ft. Hence 1,000 cu. ft. of air saturated with water vapor would contain $1{,}000 \times 0.00157 = 1.57$ lb. of moisture.

If the air is cooled to 40 deg. F. without change of pressure the volume would decrease in the ratio of the absolute temperatures or inversely as $(460 + 80)$ to $(460 + 40)$, making the volume when cooled $= 1{,}000 \times (460 + 40) \div (460 + 80) =$ approximately 926 cu. ft.

At 40 deg F. a cubic foot of water vapor would weigh 0.00041 lb., hence the quantity of water vapor that could be held in suspension in 926 cu. ft. of air at 40 deg. F. would be $926 \times 0.00041 = 0.38$ lb. and of the original moisture, $1.57 - 0.38 = 1.21$ lb. must be cooled and deposited as water at 40 deg. F.

The latent heat of the original vapor (at 80 deg. F. and pressure of 1.029 in mercury) is 1,046.7 B.t.u. and for condensing the 1.21 lb. before reducing its temperature would require $1.21 \times 1{,}046.7 = 1{,}266.5$ B.t.u.; and to cool the condensate from 80 to 40 deg. F., would require $1.21 \times (80 - 40) = 48.4$ B.t u.; for the vapor condensed out there would have to be removal of $1{,}266.5 + 48.4 = 1{,}314.9$ B.t.u.

Assuming the specific heat of superheated steam is 0.5. to cool the 0.38 lb. of vapor from 80 to 40 deg. F. would require $0.38 \times (80 - 40) \times 0.5 = 7.60$ B.t.u., so that the total heat to be removed would be 705.6 B.t.u. from air + 1,314.9 B. t.u. from moisture condensed to water + 7.6 B.t.u. for cooling remaining moisture = 2,028 B.t.u.

As one ton refrigeration per 24 hours is the removal of heat at the rate of 200 B.t.u. per min., for cooling 1,000 cu. ft. of air per min. under the conditions would require $2{,}028 \div 200$ = practically 10 tons of refrigerating capacity.

If the air is to be forced across the cooling coil at considerable velocity, the heat transmitted per sq. ft. of surface per degree temperature difference per hour will be between 5 and 20 B.t.u., depending on the velocity. Assuming that with 40 deg. exit temperature of the air and the ammonia temperature in the coil 30 deg. F. there would be an arithmetrical difference of $[(80 + 40) \div 2] - 30 = 30$ deg. and allowing transfer of 10 B.t.u. per sq. ft. of coil surface per hour, the coil surface required would be $(2{,}028 \times 60) \div (30 \times 10) = 405$ sq. ft. and if 1¼-in. pipe is used the coil would have to contain 405×2.3 ft. of 1¼-in. pipe.

Section VII

Pumps

QUESTION [Cushioning Duplex Pump]—*How is the cushioning of a duplex steam pump obtained and regulated?* W. N. R.

Answer—The steam cylinders are provided at each end with two sets of ports. Each outer port acts as the steam admission port and terminates at the end of the cylinder so as to guide the incoming steam into the clearance space behind the piston; and each inner port acts as an exhaust port and terminates nearer the middle of the cylinder. When the steam piston passes over the exhaust port, no more exhaust steam can escape and, without cushioning valves, whatever steam is thus entrapped in the cylinder is compressed and acts like a cushion in bringing the piston to rest. This compression may be excessive and, by means of hand-operated compression valves that connect the outer or steam ports of the tylinder with the inner or exhaust passages, the amount of cushioning can be reduced, by permitting more or less of the compressed exhaust to escape from the end of the cylinder chrough the steam passage into the exhaust passage.

QUESTION [Sizes of Pump Suction and Discharge Lines]—*Why are pump suction lines made larger than the discharge lines, and what determines the size of each?* W. L. F.

Answer—The pipe sizes are determined by the loss of pressure from pipe friction that is permissible when the pump is operated at fullest required capacity. For a given length of pipe the pressure required to overcome pipe friction is nearly inversely as the diameter and directly as the square of the velocity or quantity of water flowing per minute. Under ordinary conditions suction pressure is constant and is sufficient to supply the pump and overcome the pipe friction and friction of the pump passage when the suction pipe diameter is of such size as to require no greater suction water velocity than 240 ft. per min. That is, when

$$\textit{Diameter of suction pipe in in.} = \sqrt{0.1 \textit{ gal. per min.}}$$

The loss of pressure permissible in a discharge line is limited only by the strength of the pump and the power available for its operation, and as the loss of pressure in the discharge line for a given rate of pumpage is a matter of choice, the size of discharge pipe may be much less than necessary for the suction line. But under average conditions it is advisable to make the discharge pipe of such size that the velocity is not in excess of 300 ft. per min.; that is to have

$$\textit{Diameter of discharge pipe in in.} = \sqrt{0.08 \textit{ gal. per min.}}$$

QUESTION [Relative Economy of Injector and Feed Pump]—*We have a small steam power plant with an exhaust feed-water heater, and for feeding the boiler we may use an injector or a steam pump. The injector cannot be set to operate continuously, and better economy seems to be obtained from operation of the steam pump. Is this not possible?*

R. S. S.

Answer—When no exhaust-steam feed-water heater is used, an injector, while it works, is more efficient than a feed pump, because all the heat of the steam used in the injector is returned to the boiler, excepting only that portion of the heat in the steam which is converted into energy for lifting the feed water and forcing it into the boiler. With a steam pump the same amount of heat converted into energy is used for handling the water, but some of the heat is lost in overcoming pump friction and none of the heat of the steam is returned to the boiler. However, if there is trouble in starting an injector or if it "kicks back" during considerable periods when it is supposed to be operating, the steam lost by blowing to waste may easily amount to more than that required for operating a small steam pump.

QUESTION [Cleaning Injector Tubes]—*How can the tubes of an injector be cleaned of scale without injury to the injector?*

D. F.

Answer—Remove the tubes from the body of the injector and cleanse them as well as possible by washing in a strong solution of potash and hot water and after rinsing in clean hot water, dip the parts in a solution of 1 part muriatic acid to 10 parts of water, leaving them submerged only long enough to soften the deposits. The tubes then should be quickly washed in clean cold water to stop the corrosive action of the acid. After the tubes have been cleaned, if they are polished they will remain free of scale for a longer period of use in the injector.

QUESTION [Cushioning on Exhaust of Duplex Pump]—*What prevents a duplex steam pump from striking at the ends of the stroke?*

W. R.

Answer—The passage for the escape of exhaust steam from the cylinder is covered by the piston before completion of the stroke, and exhaust steam then remaining in the end of the cylinder and steam passage is compressed by further movement of the piston and acts like a cushion in bringing the piston to rest before it reaches a position to strike the end of the cylinder.

QUESTION [Sizes of Pump Suction and Discharge Pipes]—*What should be the highest velocity of water in the suction and discharge pipes of a duplex pump?*

J. E. R.

Answer—The velocity in the suction pipe should not exceed 240 ft. per min. and in the discharge pipe 300 ft. per min. The pipe sizes approximately corresponding to these velocities are given by the formulas,

$$\textit{Discharge diameter in inches} = \sqrt{0.08G}$$
$$\textit{Suction diameter in inches} = \sqrt{0.1G}$$

where G is the number of gallons pumped per minute.

QUESTION [Formula for Computing Pump Capacity]—*What is a simple formula for estimating the capacity of a pump?* W. L. A.

Answer—The capacity of a single, double-acting pump can be calculated theoretically by the formula

$$Q = \frac{a \times F \times 12}{231}$$

where Q = displacement of one double-acting plunger in U. S. gallons, a = area of plunger in square inches, and F = piston speed in feet per minute.

Where d = diameter of plunger in inches, the formula becomes

$$Q = \frac{d^2 \times 0.7854 \times F \times 12}{231} \text{ or } Q = 0.0408\, d^2 F$$

For actual capacity the displacement by the piston rod and percentage of slip of the pump must be deducted.

QUESTION [Formula for Approximate Capacity of Pump]—*What is the derivation of the formula,* $g = \frac{D^2 L N}{294}$ *for finding the number of gallons delivered by a pump?* R. O. J.

Answer—In the formula quoted,

g = Number of gallons pumped per minute;
D = Diameter of plunger or piston in inches;
L = Length of stroke in inches;
N = Number of single strokes per minute.

The plunger or piston displacement in cubic inches per minute would be area of piston × length of stroke × number of single strokes per minute, or $D \times D \times 0.7854 \times L \times N$, and as one gallon equals 231 cu, in.,

$$g = \frac{D^2 \times 0.7854 \times L \times N}{231}$$

and dividing both numerator and denominator of the second term by 0.7854 gives

$$g = \frac{D^2 \times L \times N,}{294.117} \text{ usually written, } \frac{D^2 L N}{294}$$

If N is taken to represent the number of revolutions of a single double-acting pump the result is to be multiplied by 2, and if N represents the number of revolutions of a duplex pump, which would be the same as two single pumps, the result must be multiplied by 4. It also must be borne in

mind that the formula gives results larger than actual capacity of a pump, for it makes no allowance for reduction of capacity by slippage, and in double-acting piston pumps the displacement on one side of the piston is reduced by the presence of the piston rod.

QUESTION [Horsepower of Pump]—*What is meant by the horsepower of a pump?* H. P. R.

Answer—Five different values of horsepower of a pump must be distinguished: (1) The indicated horsepower of the steam end, obtained from indicator diagrams of the steam end in the same way as determining the indicated power of a reciprocating engine; (2) the brake horsepower, or power actually transmitted to the pump, and which in a direct-acting steam pump would consist of the indicated power of the steam end less the friction of parts essential to operation of the steam end; (3) the indicated horsepower of the water end, obtained from indicator diagrams taken from the water cylinder to determine the power transmitted by the water piston or plunger; (4) the water, or developed, horsepower to express the useful work done by the pump, based on the actual weight of water discharged and total height through which the water is elevated; (5) what is commonly recognized as the nominal horsepower, which is based on the weight of water that would be displaced per minute by the plunger or piston if there were no slippage, combined with the discharge pressure plus the lift measured from the water level in the suction well to the height of the discharge gage. Thus, if the plunger or piston displacement is 1000 gal. per minute, it would be assumed that the pump actually handled $1000 \times 8\frac{1}{3} = 8{,}333$ lb. of water per minute; and if the discharge pressure indicated 40 lb. per sq. in. and the height of the gage was 16 ft. above the water level of the suction gage, the total lift would be considered as equivalent to (40 lb. $\times$ 2.3 ft. per lb.) + 16 ft. = 108 ft., making $8{,}333 \times 108 \div 33{,}000 = 27.27$ nominal pump horsepower.

QUESTION [Necessity for Air Pump with Surface Condenser]—*Why is it necessary to use an air pump with the surface condenser of a condensing engine?* H. M.

Answer—Air or other gases held in suspension in the boiler-feed water are finally discharged with the exhaust into the condenser, and in addition, atmospheric air finds its way to the condenser through leaks of the piston-rod packing and joints of connections with the condenser. Unless the gases thus accumulating in a condenser are removed as fast as received, their presence causes an increasing loss of the vacuum obtained by condensation of the steam.

QUESTION [Necessity of Auxiliary Steam Valve on Single Pumps]—*On a single-cylinder steam pump, why is it necessary to have an auxiliary steam valve?* W. C.

Answer—If the main valve of the pump were mechanically connected to the piston rod of the pump, the steam port would be covered slowly toward the end of a stroke, the stroke would not be completed and the valve would not reverse. Therefore an auxiliary valve is necessary to operate the main valve.

QUESTION [Advantage of Multi-Valves for Pumps]—*Why are many small valves used in a pump instead of a few large ones?* C. T. C.

Answer—Small valves have less lift and therefore can open and close quicker and are subjected to less wear from pounding on their seats; there is less friction, and the shorter studs or spindles are less likely to become broken from shocks or unequal distribution of pressure. A number of valves are not likely to be seated at the same time and less shock is thereby imparted to the valve deck than when a large valves comes down on its seat; and small valves are cheaper to construct and repair, and the valve seats are easier to replace than those of larger valves.

QUESTION [Relative Length of Pump Cylinders]—*Why is the water cylinder of a pump made longer than the steam cylinder?* J. P.

Answer—The water cylinder of a direct steam pump needs to be longer than the steam cylinder so there may be ample space for the water passages without obstruction by the water piston when the steam piston is at either extreme end of its cylinder.

QUESTION [Revolutions of Pump]—*What is meant by the number of revolutions per minute of a direct steam pump?* W. R.

Answer—A revolution signifies a complete cycle of the reciprocating parts, and the number of revolutions per minute would be the number of repetitions of movement of any moving part, as one of the crossheads of a duplex pump. The term is more specific for designating the number if piston reversals than "strokes," for in a single pump the term stroke is sometimes confused with double stroke, and in a duplex pump it is frequently doubtful as to what is meant by the number of strokes per minute. With the type of pump and number of revolutions given there is no ambiguity as to the number of single strokes.

QUESTION [Purpose of Dash-Relief Valves on Pump]—*Why are dash-relief valves placed on the steam end of a pump, and do they make any difference in the regulation of the stroke of a pump?* W. H. C.

Answer—Dash-relief valves are used to control the amount of opening of a communication between the exhaust passage and the regular live-steam passage, which is near the end of

the cylinder and which remains covered by the main steam valve until it reverses. The communication between the passages with its adjustable valve is simply a controlled leak to regulate the amount of cushioning and useful in slightly lengthening the stroke in case of excessive cushioning after the piston has covered the exhaust passage, especially when the pump is working slowly and with a heavy load.

QUESTION [Advantages of Modern Over Old-Style Flywheel Pumps]—*In modern designs of direct steam pumps, why it is possible to dispense with a flywheel, and what is the disadvantage of a single pump with flywheel?* R. B.

Answer—In the earlier designs of steam pumps a flywheel was necessary for obtaining momentum to reverse the direction of the stroke, and a single pump could not be started up with the crank on a dead-center or be run very slowly. In modern direct single steam pumps, the steam admission valve is steam-thrown. It is reversed at the end of the stroke by pressure of live steam under the control of an auxiliary valve that is mechanically operated by the main piston upon the completion of the stroke. In some forms of pumps the main piston and auxiliary valves are combined. In duplex pumps one engine operates the valve gear of the other; that is, reversal of one side is effected by the other side before reaching the end of its stroke. In modern pumps, besides dispensing with a flywheel for reversing, single as well as duplex pumps can be started at any point of the stroke and can be operated as slowly as desired.

QUESTION [Use of Flexible Coupling]—*Why is it necessary to employ a flexible coupling for direct connection of a motor and a centrifugal pump if both are mounted on the same baseplate?* B. F. K.

Answer—It is difficult to set the center of a centrifugal or other machine in true alignment with the center line of a motor, and still more difficult to maintain that alignment on account of difference in wear of the bearings. Hence, to obtain good running conditions when direct-connected, it is necessary to connect the motor to its load with a drag crank oı some form of coupling that will adjust itself to the difference of alignment during operation.

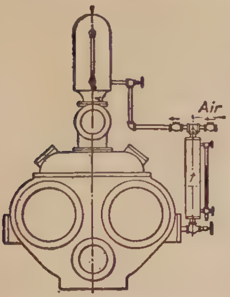
QUESTION [Weight of Water Discharged Over V-Notched Weir]—*What is the formula for the number of pounds of water at* 60 *deg. f. flowing over a* 90-*deg. V-notched weir?* T. E. S.

Answer—The formula for flow over a 90-deg. V-notch weir is cu.ft. per min. = $0.306\ \sqrt{h^5}$, in which h = the head of water measured in inches above the apex of a symmetrically placed V-notch made in "thin plate" or beveled for clearance on the discharge side of the weer. One cubic foot of

water at 60 deg. F. weighs 62.3677 lb., and for that temperature the weight of water discharged would be 62.3677 $\times$ 0.306 $\times \sqrt{h^5} = 19.08 \sqrt{h^5}$ lb. per min.

QUESTION [Loss of Air from Pump Air Chamber]—*What causes loss of air from the air chamber of a pump, and how may it be remedied?* W. T.

Answer—Water under atmospheric conditions contains about 2 per cent. air in solution and when in contact with air at higher pressure, more air is absorbed by the water. Consequently, the volume of air in an air chamber that is connected on the pressure side of a pump becomes reduced. The figure illustrates an air-charging device that utilizes the varying pressure in one of the pump chambers to compress air into the air chamber. A vertical tube t is attached to the side of the pump body and is in communication with the inside of one of the pump chambers. The upper end carries two check valves, one for suction and one for discharge, communicating with the air chamber. The pulsations in the pump chamber produce similar pulsations in the tube, thus forming an effective air compressor, without danger of the air entering the pump chamber.



QUESTION [Air Chamber for Boiler-Feed Pump]—*What should be the size and shape of an air chamber to go on a boiler-feed pump?* R. S. T.

Answer—The air chamber may be made of a large size of pipe capped at the upper end or may be of cast iron, made cylindrical or pear-shaped, the latter for greater strength of casting. The shape is immaterial so long as the vessel is strong enough to withstand the water pressure with safety and is large enough. It should have a capacity of at least six times the volume of displacement of the water cylinders, and greater volume will give smoother operation. The air chamber should be provided with a pet-cock near its lower end and another at the top for occasionally draining off the water and refilling air.

QUESTION [Power Developed in Operation of Pump]—*What number of horsepower is developed in the operation of a single, double-acting steam pump having a water cylinder* 14 *in. bore and* 18-*in. stroke of piston, making* 24 *r.p.m., pumping against* 85 *lb. pressure?* J. J. C.

Answer—Neglecting the reduction of piston displacement due to the presence of the piston rod, the water horsepower, or useful work developed would be

$$85 \times \tfrac{18}{12} (14 \times 14 \times 0.7854) \times 24 \times 2 \div 33{,}000 = 28.53 \text{ hp.}$$

The power developed by the steam end would depend on the mechanical efficiency of the pump. A mechanical efficiency of 75 per cent. would require development of $28.53 \div 0.75 = 38.04$ indicated hosepower in the steam cylinder of the pump.

QUESTION [Use of Air Chambers with Pumps]—*Why are some pumps made with air chambers while others are not?*

C. R. K.

Answer—The purpose of an air chamber is to provide an elastic element for taking up the shocks and irregularities and for inducing a uniformity of flow in the pipe line. In single and in duplex single-acting pumps, although the reversal may be instantaneous. there is of necessity a momentary stoppage that is attended by an interruption of the flow and, for smooth running, an air chamber is necessary to take up some of the water during the stroke and replenish it during the pause. An air chamber should be used with crank and flywheel pumps of any type on account of the rapid fluctuations of piston speed, but they may be dispensed with in duplex double-acing low service pumps, in small duplex pumps for general service and in small slow-running pumps such as deep-well bucket pumps and for use with centrifugal and ordinary types of rotary pumps where the flow is nearly continuous.

QUESTION [Conversion of Pump into Air Compressor]—*For best results, what changes should be made in a single direct-acting steam pump to convert it into an air compressor?*

R. W. C.

Answer—Ordinarily, the water cylinders of pumps are provided with a liberal percentage of clearance volume, as it is advantageous to their operation and adds convenience in their design and construction. For compressing air efficiently there should be a minimum percentage of cylinder clearnace and conversion of a steam pump into an efficient air compressor would at least require replacement of the water cylinder by a properly designed air cylinder.

QUESTION [Boiler Pressure Required for Pumping]—*What boiler pressure would be required to operate a direct-acting steam pump having an* 8-*in. diameter steam cylinder and a* 6-*in, diameter water cylinder, for pumping water to a height of* 100 *ft. with the suction water supplied at a pressure of* 10 *lb. per sq. in.?*

A. H.

Answer—The mechanical efficiency of a pump of the stated size, when in good working order, would be about 65 per cent. The pressure pumped against would be $100 \times 0.433 = 43.3$ lb. per sq. in. due to the head, plus the pressure required to overcome friction of the water in the discharge pipe, the latter depending on the rate of pumping and the diameter, length and fittings of the discharge pipe. Under ordinary

conditions the loss of pressure from pipe friction would be no greater than 10 lb. per sq. in., and if assumed to be equal to the suction pressure, the net resistance to movement of the water piston would be $(6 \times 6 \times 0.7854) \times 43.3 =$ 1224 lb. With 65 per cent. mechanical efficiency of the pump, the resistance to be overcome by the steam piston would be $1224 \div 0.65 = 1883$ lb., and for overcoming that resistance, the effective pressure required to be exerted on a direct-acting 8-in diameter steam piston would be $1883 \div (8 \times 8 \times 0.7854) = 37.46$ lb. per sq. in. The required boiler pressure will need to be equal to the effective pressure, plus the back pressure and the reduction of the boiler pressure by throttling in the steam pipe and steam passages of the pump. Allowing a back pressure of 4 lb. and reduction by throttling of 10 lb. per sq. in. would require a boiler pressure of $37.46 + 4 + 10 = 51.46$, or about 52 lb. per square inch.

QUESTION [Excessive Size of Pump Suction Pipe]—*Can a suction pipe of a pump be too large where the length is about* 200 *ft. and lift* 11 *ft.?* J. S.

Answer—The suction pipe should be large enough to keep the velocity of water down to about 230 ft. per min. By the use of a smaller pipe requiring greater velocity there is too much loss of pressure in overcoming pipe friction. The higher the vacuum required at the pump the greater the amount of air liberated out of the water, giving trouble in operation of the pump. On the other hand, when a larger suction pipe is used, there may be less pipe friction from lower velocity, but more time is given for liberation of air from water that is subjected to less than atmospheric pressure. However, the pipe cannot be too large for those portions where the pressure in the suction pipe is above the pressure of the atmosphere.

QUESTION [Size of Pump-Suction Valves]—*What would be the size of suction-valve areas of a steam pump for operation at a piston speed of* 100 *ft. per minute?* G. L. J.

Answer—The water velocity through the valve-seat open areas is generally taken as 222 ft. per min., which gives a valve area 45 per cent of the plunger area, with piston speed of 100 ft. per min. Greater liberality in valve area is conducive to good service and durability, and specifications for water-works pumps frequently demand clear valve areas 50 to 60 per cent. of the plunger areas.

QUESTION [Compressed Air No Assistance to Suction Lift]—*Would not compressed air, introduced at the top of a capped well casing, be of assistance in raising water with a pump suction pipe that is let down in the casing?* J. H.

Answer—The compressed air will simply depress the level of the water in the casing without any effect of raising the level of the water in the pump suction pipe, for it would con-

tinue to depend on the head and flow of the ground water. If the air pressure were sufficient to depress the water down to the foot of the casing, it would prevent any water from entering the casing.

QUESTION [Velocity in Pump Suction and Discharge Pipes]—*What velocities are allowable for water pump suction and discharge pipes?* G. L. J.

Answer—In order that loss of pressure from pipe friction may not be too great for ordinary lengths of pipe lines, the velocity of water in the suction pipe should not exceed 240 ft. per minute and that in the discharge pipe 300 ft. per minute.

QUESTION [Regulating Pump with Variable Steam Pressure]—*Our boiler pressure varies from* 80 *to* 100 *pounds and we are desirous of maintaining a regular speed of a tank pump. Can it be accomplished with a pressure reducing valve or a pump governor?* D. A. H.

Answer—If there must be a uniform number of strokes per minute it will be necessary to control the pump with some type of centrifugal throttling governor driven from aconnecting rod and crank-shaft motion. But for controlling the supply of steam so the speed of the pump will be adapted to maintaining a constant discharge pressure, use a pump governor that is operated by the discharge or tank pressure.

QUESTION [Cause of Pump Becoming Airbound]—*What causes a pump to become airbound, and how can the trouble be remedied?* S. P.

Answer—Under atmospheric conditions water contains about 2 per cent air in solution. When subjected to less pressure the air expands, and when given sufficient time, as while the water is flowing through a long suction pipe, the air becomes liberated from the water, usually in the form of collections of small bubbles that adhere to the pipe or pump surfaces, or the air gathers in pockets of the pump spaces. The trouble can be prevented by increasing the head of the suction supply and by running the pump slower, so the pump cylinder will be filled with less reduction of pressure of the suction water.

QUESTION [Height of Pumping Water]—*A direct-acting steam pump with steam piston* 12 *in. diameter and water piston* 8 *in. diameter is operated with steam at* 110 *lb. per sq.in. gage pressure. To what height in feet can the pump raise water, assuming* 70 *per cent. efficiency?* G. G. M.

Answer—If the total pressure exerted on the steam piston is transmitted to the water piston, 110 lb. per sq.in. gage pressure acting on the steam piston, opposed by back pressure of the atmosphere, would exert a pressure of $(12^2 \div 10^2) \times 110 = 158.4$ lb. per sq.in. on the water piston; one foot head of water exerts a pressure of 0.433 lb. per sq.in.,

and without friction of water in the pump or pipes the pump could raise the water to a height of $158.4 \div 0.433 = 365.8$ ft. above the level assumed by the suction water under atmospheric pressure. With 70 per cent. efficiency the height would be 0.70 of $365.8 = 256$ feet.

QUESTION [Glass Gage for Pump Air Chamber]—*What should be the height of a glass gage on an air chamber of a pump that operates against a gage pressure of* 105 *lb. per sq.in.?* J.W.C.

Answer—With a discharge pressure of 105 lb. gage or about 120 lb. per sq.in. absolute, the air contained in the air chamber at atmospheric pressure would be compressed to about $\frac{15}{120}$ or $\frac{1}{8}$ of its original volume. If the air chamber is a vertical cylinder, the water level would be raised to within about $\frac{1}{8}$ the height measured from the top. With a pear-shaped air chamber having the lower or conical portion about 3 times as long as the diameter of the hemispherical top, the water level would rise to within about $\frac{1}{3}$ of the diameter below the top of the air chamber. For indicating the conditions, it is well to have the glass gage connections so arranged as to show any standing of the water level from the top down to about one-half the total depth of the air chamber.

QUESTION [Jumping of Feed Pump]—*What causes a direct steam feed pump to make short jumps of the plunger at the beginning of the stroke?* E. M.

Answer—The jumping indicates presence of air and may result from running the pump too fast, or the pump design may be such that the water is not always rising and there are pockets where air can accumulate.

QUESTION [Lame Running and Jumping of Pump]—*What would cause a feed pump to run lame on the instroke and start with a jump on the outstroke?* P. J.

Answer—The trouble may be caused by a defective suction valve that fails to open fully during the instroke and is not tightly closed during the outstroke.

QUESTION [Trouble Pumping Hot Water Under Head]—*What causes trouble pumping hot water when the water flows by gravity to the pump?* H. F. J.

Answer—If the piston displacement of the pump is faster than flow of the suction water can supply, there will be cavitation in the pump cylinder and suction passages to the pump in which the pressure becomes so reduced that the hot water bursts into steam of pressure corresponding to the temperature and the pump cylinder thereby becomes steambound. The remedy is to stop the pump, open all the drain valves on the water end of the pump and leave them open until the water flows free of vapor, and then start and run the pump slowly.

QUESTION [Trouble from Long Pump Suction Pipe] —*Trouble is experienced in operating a* 6 *x* 7 *x* 12-*in. light service pump on account of leakage in the* 4-*in. suction pipe, which is* 1300 *ft. long. The suction pipe is buried* 4 *ft. below the surface of the ground. What would be the best way to locate the leak?* F. B. S.

Answer—A tight suction pipe, of the length stated, is likely to become airbound in a short time from liberation of air out of the water during the great length of time the water is traversing the pipe while under less than atmospheric pressure and that trouble is commonly mistaken for a leak or solid stoppage of a long suction pipe. Before assuming that a leak has developed, it would be well to stop the entrance end of the pipe and use water from a tank under pressure or employ the pump for testing whether the pipe is tight against pressure. If it is found to be pressuretight, the air "plugs," which usually consist of accumulations of air bubbles, will have to be pushed out of the pipe by forcing water through it. If the pipe is found to have a leak, its location can be determined by making pressure tests of different sections, beginning, for instance, at the middle of its length and testing each half, thus narrowing down the search to a small section of length that contains the leak. In most soils where the leak is considerable, its location will become revealed on the surface of the ground by keeping the pipe under water pressure for 12 to 24 hours.

QUESTION [Hammering of Geared Feed Pump]—*We are troubled with hammering of a geared triplex pump used for feeding hot water that is supplied from an open tank under a head of about three feet. Where should we look for the cause, and how can it be remedied?* W. L.

Answer—It is probable that the pump is driven too fast for the pump chambers to become filled on account of length of suction pipe or small size of pipe and passages, which results in slippage of the pump on the suction strokes and hammering when the plungers come down on water that fills only a portion of the cylinders. If the pump is belt-driven and it is not otherwise convenient to try it out with slower speed, note whether the hammering stops when the speed is slowed down with the belt slipping from being thrown partly on the loose pulley.

QUESTION [Water Discharged from Orifice Under Different Heads]—*If there are two openings of the same size in the side of a standpipe, one* $6\frac{1}{4}$ *ft. and the other* $56\frac{1}{4}$ *ft. below the surface of the water, what would be the relative rates of discharge from the orifices?* C. V. J.

Answer—The theoretical velocity of discharge of water through an oriffice is the same as the velocity acquired by fall of a body in vacuo, given by the formula, $v = \sqrt{2gh}$, in

which v represents the velocity in feet per second, g the acceleration due to gravity, usually taken as 32.16, and h the height of the fall in feet, which, in considering the velocity of discharge through an aperture, usually is taken as the height in feet measured from the center of the aperture to the surface of the liquid. The actual velocity of discharge if given by the formula $V = c\sqrt{2gh}$, in which c is a constant less than unity, its value depending on the size, shape and form of the sides of the orifice and the influence which the sides of the standpipe or vessel that contains the liquid may exert on the velocity of approach to the orifice. If the orifices are small compared with the size of the standpipe and depth of water below the lower orifice, the coefficient will be substantially the same for orifices of identical size, shape and form of sides. Hence if D is the distance in feet from the top of each orifice to the center, and one orifice is 6.25 and the other 56.25 ft. below the surface, the relative velocities and rates of discharge would be as $\sqrt{6.25 + D}$ to $\sqrt{56.25 + D}$.

QUESTION [**Operation of Reciprocating and Centrifugal Pumps in Parallel**]—*Can a centrifugal and a reciprocating pump be operated in parallel for boiler feeding?* J. K.

Answer—When a centrifugal discharge is connected into a feed line through a Y-branch, it can be operated in conjunction with a reciprocating pump discharging into the same feed line provided the reciprocating pump is fitted with an air chamber that prevents pulsation of the discharge and is not operated fast enough to maintain a feed-pipe pressure higher than the pressure that can be built up by the centrifugal. As the action of the reciprocating pump will be positive, the delivery by the dentrifugal will not be parallel in the sense of bearing a constant ratio, but will be the additional flow that can take place through the check valve and the feed-pipe connections at the highest pressure that can be built up by the centrifugal.

QUESTION [**Paralleling Discharge of Centrifugals**]—*We operate two centrifugal units for supplying separate 16-in. service mains of a town and a mill water supply. The discharge of each centrifugal has a branch with check valve to each service main. It is found that both pumps cannot be made to discharge at the same time into the same main, as one will build up higher pressure than the other. Could not this trouble be overcome by removing the flaps from the check valves?* H. A. T.

Answer—Closure of a check valve is an effect and not a cause of back pressure, and when a check valve becomes seated, somewhat greater intensity of pressure is required to raise the flap than that holding the valve to its seat, the difference depending upon the design and size of the valve. To operate in unison, each pump must have a building-up capacity in excess of the margin necessary for opening the

check valves, and to insure discharge from both centrifugals at the same time it is necessary to limit the discharge to the highest pressure that can be overcome by the weaker centrifugal, less the extra pressure required for opening the check valve on its discharge. The built-up pressure of either pump can be reduced to the selected standard by chopping down its discharge or by returning part of its discharge back to the suction. To save constant attention of an operator this can be accomplished by a discharge-pressure regulating valve, or by an automatically operated bypass valve.

QUESTION [No Advantage Raising up Submerged Centrifugal]—*A centrifugal pump is operated submerged for raising water to a height of 75 ft. above the pump. Will the head pumped against and power required be reduced by raising the pump 8 feet?* W. T.

Answer—The pump must operate against a pressure which is equal to the pressure due to the head above the pump, plus pipe friction, minus the pressure due to the head of the section water above the pump. This difference would be unchanged by raising or lowering the pump, excepting in the small difference of pipe friction, due to variation in length of discharge pipe. Therefore, raising the pump and pumping at the same rate would require practically no less power for raising the water to the present elevation above the level of the suction water.

QUESTION [Economy of Direct Steam Pumps]—*What is the average steam consumption per horsepower-hour of small duplex boiler-feed pumps of sizes such as 6 x 4 x 6 and 8 x 5 x 10, when operated with steam at 100 lb. pressure?*

L. S. J.

Answer—The steam consumption of direct-acting steam pumps if necessarily very high, as steam must be admitted to follow full stroke, the piston speed is low, the percentage of cylinder clearance is much greater than in ordinary steam engines and generally such pumps are operated against high back pressure. All these conditions are more detrimental to economy in the smaller sizes of pumps, and for ordinary sizes used for boiler feed pumps, such as 6-in. x 4-in. x 6-in. or 8 in. x 5-in. x 10-in., direct-acting pumps have steam efficiencies less than one-tenth that of a good engine; that is, they use from 150 to 200 lb. of steam per indicated horsepower per hour. But where the exhaust can be utilized for heating boiler-feed water, or displaces the expenditure of live steam for heating purposes, the excessive steam consumption per horsepower-hour becomes a negligible consideration.

QUESTION [Sizes of Pump Suction and Discharge Pipes]—*What should be the highest velocity of water in the suction and discharge pipes of a duplex pump?* J. E. R.

Answer—The velocity in the suction pipe should not exceed 240 ft. per min. and in the discharge pipe 300 ft. per min. The

pipe sizes approximately corresponding to these velocities are given by the formulas;

$$\textit{Suction diameter in inches} = \sqrt{0.1G}$$

$$\textit{Discharge diameter in inches} = \sqrt{0.08G}$$

where G is the number of gallons pumped per minute.

QUESTION [Increasing Cushion on Duplex Pump] —*How can the amount of cushioning be increased on a duplex pump that is not provided with cushion-regulating valves?*

A. R. D.

Answer—The exhaust can be closed earlier, and thereby provide for a longer cushion, by turning off the diameter of the piston sufficiently to allow for a piston ring with an offset overlapping the piston. In this way a piston ring can be used that will be as wide as the piston, and a cushion can be obtained as long as the piston will permit.

QUESTION [Resetting Spool on Duplex Pump]—*On a duplex pump how would the proper position be found for resetting a crosshead spool that had become loose on one of the piston rods?*

H. G.

Answer—When the spool is properly set, the rocker arm which it carries will be at right angles with the piston rod when the piston rod that carried the spool is in the middle of its stroke. To place the piston rod in the middle of its stroke, move the steam piston as far as it will go toward the head end of its stroke, and after making a mark on the piston rod at the end of the stuffing-box gland, move the piston as far as it will go in the other end of the cylinder and make a similar mark on the rod. After locating a mark halfway between those marks, move the piston and rod so the middle mark will be at the end of the stuffing-box gland and, with the piston thus placed in the middle of its stroke, set the crosshead to such a position that the rocker which it carries will be at right angles with the piston rod.

QUESTION [Pressure at Discharge Coupling of Sinking Pump]—*What is the pressure on the pump couplings of a 3-in. rubberlined hose, delivering* 100 *gal. of water per minute from a mine sinking pump against a head of* 275 *ft.?*

S. A. M.

Answer—Water pressure due to a head of 275 ft. would be equivalent to $275 \times 0.433 = 119$ lb. per sq. in., and for raising the water the pressure discharged into the lower end of the discharge line will be that pressure plus the pressure required for overcoming friction plus an excess pressure due to pulsation of the pump. The pressure required for overcoming friction to discharge through hose depends on the kind and roughness of the lining, freedom from kinks or short bends, the length of the hose and the rate of discharge. For smoothest surface of rubber lining and straight

hose, discharging at the rate of 100 gal. per min., the frictional resistance will amount to only about one pound per sq.in. per 100-ft. length of hose. The pressure due to pump pulsations depends on the number of piston reversals per minute and the smoothness of operation of the pump. The pressure effect of such impacts cannot be computed, but under ordinary conditions would not exceed 20 lb. per sq.in. With delivery at the rate of 100 gal. per min. through 300 ft. of straight 3-in. smooth-lined rubber discharge hose, the maximum pressure at the pump coupling would be about 142 lb. per sq.in., but with a kinked or rough-lined hose there would be a rapid increase of frictional resistance and of total pressure required to effect a stated rate of discharge.

QUESTION [**Omission of Counterbores from Duplex Pumps**]—*Why are not the cylinders of a duplex pump counterbored?* S. O. R.

Answer—The cylinder of a reciprocating engine that is constrained to a uniform length of stroke is counterbored so the piston may override the ends of the main bore and thus prevent the wear from forming shoulders in the bore at each end of the stroke. With a duplex steam pump the absolute length of stroke and portion of the cylinder over which the pistons travel vary with speed, steam pressure, valve setting and chance amount of "lost motion" of the valve gear; there is little likelihood of formation of abrupt shoulders from wear and, for ordinary duplex pumps, there is little or no advantage in counterboring the cylinders.

QUESTION [**Steam Cylinders for Compound Duplex Pump**]—*A compound duplex steam pump of 12-in. stroke and having 8½-in. water cylinders is to work against a discharge pressure of 180 lb. per sq.in. What should be the sizes of the high- and low-pressure steam cylinders if exhaust takes place against 2 lb. pressure above the atmosphere?* G. R. W.

Answer—With ordinary clearance and other proportions of design, each side of the pump should have a 9-in. diameter high pressure and 15½-in. low-pressure steam cylinder.

QUESTION [**Feasibility of Using Air Lift**]—*We used two duplex pumps for raising water from three driven wells that are 110 ft. deep. The pumps are located in a pit that is about 10 ft. deep, but the water level in the wells is so low that the suction is 25 to 28 in. vacuum and there is considerable pump slippage. Would it not pay to install an air lift to raise the water to the pit and pump from the pit?* G. H. D.

Answer—The conditions are favorable to use of an air lift and as it is quite evident that by pumping, the level of the well water drops beyond the suction lift of the pumps, the pumps must be lowered or some type of deep well pump should be installed. By submergence of air lifts within about 3 ft. of the bottoms of the wells, there would be fair air lift effi-

ciency for delivery to the pump pit, and nearly as good efficiency if the air lifts discharge two or three feet above the surface of the ground.

QUESTION [Discharge of Attached Circulating Pump]—*What number of British gallons per minute would be handled by an attached double-acting circulating pump* 15 *in. in diameter by* 21 *in. stroke, with* 2-*in. piston rod, driven by a beam from the crosshead of an engine making* 80 *r.p.m., assuming that the pump barrel is filled* 80 *per cent at each stroke?*

C. E. G.

Answer—The 15-in. diameter piston would have an area of 15 × 15 × 0.7854 = 176.71 sq.in., and with a 2-in. diameter piston rod the net area of the piston on the rod side would be 176.71 − (2 × 2 × 0.7854) = 173.57 sq.in. For a 21-in. stroke, 80 per cent of the pump barrel filled at each stroke, and for 80 r.p.m. of the engine, the water displacement by the pump would amount to (176.71 − 173.57) 21 × 80 per cent × 80 = 470,776.322 cu.in. per min. As a British gallon is 277.274 cu.in. the quantity of water handled would be 470,776 ÷ 277.274 = 1,697.8 British gallons per minute.

Section VIII

Elevators

Section VIII

Elevators

QUESTION [Elevator Goes up with the Controller on the Down Position]—*What causes an elevator to take an upward motion when the controller handle is pushed over to the down-motion position?* A. A. A. R.

Answer—If the wiring is correct, the cause of an elevator taking an upward motion with the controller in the down position will depend upon the type of machine. If the motor and controller have been disconnected and then reconnected incorrectly, then, of course, the trouble is in the wiring. Assuming that the wiring has not been interfered with and that the machine is of a semi-magnetic control type with a mechanical brake, then there are two conditions that could cause the trouble on a spur-geared type machine. First, with the car over-counterweighted, the operator may pull the cables to a position where the brake is released, with power not cut into the motor. In that case the counterweight would be free to run the car in the up direction. Under similar conditions, but with the power circuit open, if the operator pulls the car cable to the down-motion position, the brake would be released, but there would be no power to the motor to drive the car in the position it should go, and it would be carried upward by the counterweights going down. These conditions would also be true for a worm-geared machine, but in this case the friction of the worm gear is generally sufficient to prevent movement of the car. If the car were loaded to a point where it was heavier than the counterweight, it would take a downward instead of an upward motion under the foregoing conditions.

In the case of an elevator with a magnet brake, the only case where the conditions referred to in the question could apply would be where the motor circuit was opened in such a way that power could not be supplied to the motor, but the brake coils could be energized and release the brake. In such a case on a spur-geared machine, where the counterweights are heavier than the car and its load, the car would move in an upward direction. However, this condition is remote.

QUESTION [Interlocks on Elevators]—*On elevators what are the interlocks and for what are they used?* W. H.

Answer—Interlocks are mechanical devices that require that the elevator door be closed before the car can be started

in either direction. They require that the car be at or within a predetermined distance above or below the landing before the door can be opened. There are also electrical devices, sometimes referred to as interlocks, which are intended to have the same effect. It is important that there be a means through which the car operator can render the locking device inoperative in case of emergency, and the device should be so arranged that evidence of the operator's or other attendant's act in this respect will be easily noticeable. A desirable arrangement is a mechanical interlock on the shaft door and, in addition, a car door that closes with this door. The car should be inoperative until both the car door and the enclosure door are closed and latched, and the opening of either door should be impossible until the car is brought to rest within two inches above or below the landing.

QUESTION [Putting Air in Pressure Tanks on Hydraulic Elevators]—*On hydraulic elevators how is the air supply maintained in the pressure tank and what is its purpose?*

B. W.

Answer—Air is introduced into the pressure tank either by an independent air compressor or by throttling down the suction valve on the water pump and opening an air-intake valve. Where there are centrifugal pumps with considerable slippage due to wear, and where there is violent agitation of the water in the surge tank, sufficient air may enter the suction to keep the system supplied. In fact, such a system sometimes has to be relieved of air. The function of the air is by its expansive force to cause water to flow from the tank to the cylinder when the controlling valve is open. Water alone will not do this satisfactorily because it is practically incompressible. A combination of the two is necessary and is ideal for performing this function. To pump directly into the elevator cylinder would require the use of equipment much larger than is otherwise necessary. Moreover, even if the pump were of the centrifugal type or had a great many cylinders, the car would not ascend with a smooth motion.

QUESTION [Elevator Car Miles per Day]—*What mileage will an elevator make in the average building?*

Answer—Car miles per day for each elevator in an ordinary small Federal building are taken at 5; while in the large buildings of six or eight stories, in places like New York and Philadelphia, and in the departments at Washington, each passenger elevator will average 10 car miles per day.

Kilowatt-hours per car mile for electric elevators equal 5.

Steam per car mile for hydraulic elevators operated by compound duplex steam pumps equals 1,000 lb., while for high-duty pumps it amounts to 650 pounds.

QUESTION [Valves on Hydraulic Elevators]—*How many valves should there be in the control of a hydraulic elevator and for what are they used?* M. E. W.

Answer—The horizontal type of hydraulic elevator generally has four valves—the pilot valve, the control valve, the automatic stop valve and the vacuum breaker. Vertical-cylinder hydraulic elevators generally have five valves and sometimes six—the pilot valve, the control valve, the automatic stop valve, the vacuum breaker, the governing valve in the circulating type, and in some instances a cushioning or relief valve in the lower end of the circulating pipe. Some plunger-type elevators have three valves—the pilot, control and limit. Others have the pilot valve, composed of two separate parts; the control valve, also of two separate parts, and an automatic limit valve for the upward travel and a similar one for the downward travel.

The object of the pilot valve, which is controlled directly from the car lever, is to cause the main or control valve to operate. This latter valve controls the flow of water from and to the cylinder. The function of the automatic stop valve is to control automatically and independently of the control valve the flow of water from and to the cylinder at terminal landings. The vacuum breaker prevents a vacuum from forming in the discharge pipe after the control valve is brought from the down position to the stop position.

The governing valve in the circulating pipe is to prevent overspeeding of the car and has the effect of regulating the area of the circulating pipe. The stop valves in the piping should be of the gate type with a bypass, and used only when it is desired to shut off all water supply to and from the elevator engine. These stop valves under no circumstances should be used for regulating the speed of the elevator. With some types of elevator and common conditions of loading and counterweighting, the throttling down of the discharge stop valve would prevent the control valve from returning to its stop position when the car is descending. The throttling down of the stop valve on either the pressure or the discharge pipes is a frequent cause for unsatisfactory action of the control valve. Stop valves should be full open, less one revolution, during normal operation.

QUESTION [Calculating the Pressure in Hydraulic Elevator Cylinders]—*How is the pressure required in the cylinder of a hydraulic elevator of the multiplying-sheave type, used for passenger service, determined?* W. H.

Answer—On any hydraulic type having multiplying sheaves, the calculation of approximately the maximum pressure necessary may be made as follows:

A. Assume a live load of 100 lb. per sq. ft. of effective car-

floor area. This weight is arrived at by taking the average weight per person at 150 lb.; minimum space occupied 14x14 in. or 196 sq in., equals 0.765 lb. per sq. in. or 110 lb. per sq.ft.

B. Multiply the square feet of effective car-floor area by 110 to get the total live load.

C. To the live load add the unbalanced weight of the empty car to obtain the maximum load to be lifted.

D. Multiply the load to be lifted by the ratio of piston travel to car travel.

E. For a 10 to 1 reduction increase the product of "D" by 60 per cent, for 8 to 1 by 50 per cent, for a 6 to 1 by 40 per cent, for a 4 to 1 by 30 per cent, and for a 3 to 1 by 20 per cent. These percentages are arbitrary and vary more or less, but are correct enough for practical conditions. They represent the loss through friction.

F. After obtaining the result under "E" divide it by the square inches of cylinder area. The quotient will be the pounds gage pressure required on the system to start a load properly. The pressure required to continue the load in motion is much less than that required to start it.

As an example assume a horizontal-cylinder pushing type, 10 to 1 reduction elevator machine with a 26-in. diameter cylinder:

A. Car floor 5.5×5 ft. = 27.5 sq. ft. area.

B. 27.5×110 lb. = 3,025 lb. live load, to which must be added,

C. Say 1,000 lb., which makes a total of 3,025 + 1,000 = 4,025 lb. load to be lifted.

D. Multiply the value of "C" by the gear ratio, or 10, the product being 40,250. Increase this by

E. 60 per cent, which gives a total load of $40{,}250 \times 1.60$ = 64,400 lb., representing the live load, the unbalanced weight of the car, the frictional loss and effort to first overcome gravity.

F. Divide "E" by square inches of cylinder area, which equals $26 \times 26 \times 0.7854 = 531$. This gives $64{,}400 \div 531$ = 121 lb. gage required in the cylinder.

The automatic regulators on the pumping system should be set for 120 lb. and the safety valves set for not over 130 lb.

UESTION [Action of Governors on Electric Elevators]—*What is a governor in connection with a car safety device on an electric elevator? Is there more than one kind of governor? How is the safety device actuated when the car overspeeds?*

W. H.

Answer—Centrifugal-weight governors are generally used on traction elevator machines and on most other modern machines, but a large number of side-cam governors are also in use. The weight governor is so arranged with shafting,

gears and cams that the centrifugal force, when the car overspeeds, causes the weights to leave their normal position, and this has different effects with different governors. On the high-speed traction elevator the governor first short-circuits a portion of the shunt-field resistance, but if the car continues to overspeed, the governor next opens the circuit through the potential switch, which disconnects the motor from the line. If for any reason the car should not instantly slacken speed, a tripping device will be actuated and suitable jaws will catch the governor cable, stop it and cause the safety device under the car to grip the guide rails and stop the downward motion of the car. Probably the most reliable of all types of governors under all conditions of operation, repair, adjustment, cleanliness and general maintenance is the type where the weights revolve with a vertical spindle or shaft. Sidecam types and weight types, when the latter are mounted on a horizontal shaft, are not so reliable, except under ideal conditions of maintenance. The function of the governor which is equipped with an electrical switch is, first, to open the switch should the car overspeed, and second, to cause the gripping of the governor cable. The final function of any governor is to cause the gripping of the governor cable.

QUESTION [Multiplying Sheaves on Hydraulic Elevators]—*What are the positions and functions of the multiplying sheaves on hydraulic elevators?* M. E. W.

Answer—On the vertical-cylinder type machine one or more sheaves are located on the traveling crosshead to which the piston rods are attached. At some point in the shaftway above these are mounted one or more stationary sheaves. The hoisting cables start from a fixed point in the shaftway, travel downward to the lowest sheave on the crosshead, upward to the lowest stationary sheave, downward to the next lowest sheave on the crosshead, upward to either another stationary sheave or to the overhead sheaves, as the case may be. From the overhead sheaves they travel downward to the car. On the horizontal pushing type one set of sheaves is mounted on the closed end of the cylinder. One end of the hoisting cables is fastened at a dead point, consisting of a lug on the cylinder head. From this point they travel first around one traveling sheave, starting over its top, then around a stationary sheave, to another traveling sheave, to another stationary sheave, continuing around all the sheaves, and from the last stationary sheave they travel upward to the overhead sheave and downward to the car. The function of the multiplying sheaves is to secure a length and speed of car travel in excess of the length and speed of the piston travel, it being impracticable to have the piston travel equal to that of the car, except in the plunger-type machine, where the cylinder goes down in the ground to a distance approximately equal to the car travel.

Section IX

Valves

Section IX

Valves

QUESTION [Blind of a Valve]—*What is meant by a valve being one-eighth inch blind?* C. T.

Answer—The term blind is sometimes used with reference to a valve to signify that it is closed. One-eighth inch blind would signify that the valve overlaps the opening edge of the port one-eighth inch.

QUESTION [Object of Double-Ported Engine Valves] —*What is the object of providing an engine with double-ported valves?* A. H.

Answer—The purpose of suing a double-ported valve is to obtain the desired area of opening with half as much movement of the valve, thereby reducing the loss from friction and wear of the rubbing surfaces and securing the advantages of a single-ported valve with twice the length of port opening.

QUESTION [Pressure-Reducing and Pressure-Regulating Valves]— *'hat is the difference between a pressure-reducing and a pressure-regulating valve?* R. E. T.

Answer—Any valve that effects a reduction of pressure, as by throttling the flow of a gas, vapor or liquid, would properly be called a pressure-reducing valve; and a valve that limits the discharge to a constant maximum of quantity or of pressure would be a regulating valve. A valve having the purpose of reducing steam of boiler pressure to a constant lower pressure suitable for steam heating would properly be designated as a pressure reducing and regulating valve.

QUESTION [Overtravel of Valve]—*What is meant by overtravel of an engine valve?* R. L.

Answer—The overtravel of a valve is the distance the outside edge of the valve passes by the inside edge of the port; or in other words, the amount of movement that takes place in excess of the movement necessary for completely uncovering the port.

QUESTION [Ball Versus Disk Check Valves]—*What are the relative merits of ball and of disk check valves?* H. D. Y.

Answer—The principal advantages of ball check valves are simplicity and compactness and durability for situation where leakage is not of serious importance. For a ball check to remain tight with use, the ball must continue in the form of a perfect sphere and the wear of the seat from pounding must

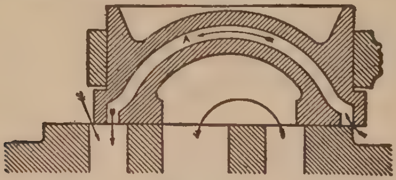
shape the seat to the radius of the ball. The seat may continue round, but unless its surface conforms to the spherical form of the ball, contact between the valve and seat is reduced to a line and there is no breadth of seal against leakage. To obtain a good bearing surface for the seat and correct spherical form of ball requires a high order of mechanical skill. Disk valves tend to conform to their seats from use, they are readily repaired in place and therefore are preferable for situations where check valves must be tight and are accessible for repairs.

QUESTION [Safety Valve for Double-Grate Down-Draft Boiler]—*In selecting the size of a safety valve in accordance with the size of grate, how should the grate area be estimated for a double-grate down-draft boiler?* B. R.

Answer—Where a boiler is provided with a double grate for down-draft combustion, the grate area taken for estimating the size of safety valve should be the area of the lower grate plus one-quarter of the area of the upper grate.

QUESTION [Trick's Slide Valve.]—*What is the difference between a Trick valve and an ordinary D slide valve?* F. A.

Answer—The main difference is that Trick's valve affords double admission. Live steam is admitted past the outer edge of the valve just as with the ordinary *D* slide valve and



additionally through a passage *A* in the valve, as shown in the sketch. When the outer edge of the valve uncovers the port for admission, the port of the passage *A* at the opposite end of the valve is also uncovered, thus giving a double admission of steam, the same as a double-ported valve. This valve does not, however, give double exhaust, as the passage *A* is not opened to the exhaust.

QUESTION [Compression Regulating Valves of Duplex Pump]—*What is the purpose of compression valves of a duplex steam pump and how are they connected?* F. C.

Answer—In ordinary duplex steam pumps, at each end of the steam cylinders, the opening of the exhaust passage into the cylinder wall is farther from the end of the cylinder than the opening of the steam passage. The latter usually is at the very end of the cylinder bore or wholly or partly in the cylinder head. Compression of exhaust steam is effected by

the piston covering the exhaust passage before the exhaust stroke has been completed. Should the piston cushion too heavily, the compression can be reduced by opening a valve in a connection made between the end of the cylinder and the exhaust passage or, as commonly done, by regulation of a valve that controls the amount of opening of an aperture in the partition between the steam and exhaust passages. When the main valve shifts for reversing the pump, the exhaust port is covered at the same time the steam port is uncovered, so that during the return stroke of the piston the escape of live steam past the compression-regulating valves is as effectually prevented as from the cylinder itself.

QUESTION [Setting Slide Valve Without Uncovering]—*How can the valve of a slide valve engine be set without uncovering the valve chest?* W. R.

Answer—The valve can be set approximately without uncovering the valve, from judgment of the equality of flow of steam from the cylinder waste cocks for determining when the steam ports are nearly covered by the admission edges of the valve. With the cylinder waste cocks wide open and the throttle valve opened only a very little, place the crosshead so the piston will be at a point of the stroke first in one end and then in the other end of the cylinder to obtain a very small but apparently equal discharge of steam from the waste cocks, and for each position, make a mark on the valve rod at the gland of its sufflug-box and locate a mark midway between those marks. Then turn the engine over slowly and mark the valve rod in the same manner when it is farthest in and out of the stuffing-box and locate a mark midway between these marks. If the first and second middle marks do not agree, it is because the valve travels more to one side than to the other side of the center of the ports. If the mark that represents the middle of the valve travel falls on the steam chest side of the first middle mark, the distance between the marks shows the amount that the valve rod is too long, or falling on the crank side, the distance shows the amount the rod is too short for giving symmetrical travel of the valve, and the difference should be rectified by adjustment of the length of the valve rod or of the eccentric rod. After thus equalizing the valve travel, place the engine on a center and turn the eccentric on the shaft in the direction of rotation of the engine and set the eccentric on the shaft when discharge of steam from the waste cock indicates that the valve is beginning to open and determine whether the same opening of the valve has been obtained when the engine is placed on the other center.

QUESTION [Repairs to Leaky Piston Valves]—*How can leaky piston valves be made tight?* R. D.

Answer—If leakage is from wear or cutting of the valve seats they will need reboring and in nearly every case the

packing rings will need to be replaced, for although the old rings may show no signs of cutting, they will be worn to shapes that will not conform to the rebored seats.

QUESTION [**Setting Slide Valve With Lead**]—*What is the method of setting the valves of an upright slide-valve engine and how much lead should there be?* D. M.

Answer—The method is practically the same as for setting the valve of a horizontal slide-valve engine. Remove the steam chest cover and, with the eccentric fast on the shaft, turn the engine over by hand and find whether the admission edge of the valve overrides the admission edge of the steam port as much for one of the cy inder as for the other. If the amounts are not the same, they should be equalized by adjustment of length of the valve rod. Then place the engine on a center and set the eccentric to such a position that the valve will give the desired lead opening at the beginning of the stroke, but with the eccentric so placed with reference to the crank that the opening will be increased by turning the engine in the running direction. Next, place the engine on the opposite center and if the lead is not the same, obtain one-half the desired lead by readjustment of the valve rod and one-half by readjustment of the eccentric, repeating these adjustments on one end and then the other until the lead is equalized. The appropriate amount of lead opening depends on the size, design and speed of the engine, and also on the steam pressure and the load to be carried. The lead should be at least enough to obtain full steam-chest pressure in the cylinder at the beginning of the stroke when the engine is running at normal speed, and also as much as may be necessary, in conjunction with compression of the exhaust, to insure sufficient cushioning of the piston for quiet running of the engine.

QUESTION [**Valve Cutting from Lack of Lubrication**]—*What effect would it have on the steam valves and valve seats of a tandem air compressor, with 7- and 14-in. compound steam cylinders, using the compressor by admitting live steam through a bypass to the low-pressure cylinder?* J. C. R.

Answer—A bypass to the low-pressure cylinder is usually intended only for warming up and starting, and if lubrication is not provided during extensive use of the bypass for operating the compressor, the valves and valve seats are likely to become cut.

QUESTION [**Damage from Water in Cylinder**]—*Would breaking of an engine cylinder from presence of water be more likely to occur with a D slide valve or with a piston valve?*

G. R. W.

Answer—A "smash" is more likely with a piston valve than with a flat D slide valve because, with a piston valve, the only relief passage for the water is the chance amount of port opening at the time the piston compresses the water in the

cylinder; whereas, a D slide valve can be forced away from its seat and the water can be vented to the steam chest.

QUESTIONS [Objections to Excessive Blowdown of Pop Valve]—*Why is it more objectionable to have a boiler safety valve blowdown 10 lb. than 4 lb. below the pressure for which the valve is set to pop?* R. W.

Answer—The greater the blowdown the lower must be the available working pressure below the safe working pressure for the boiler. In addition to this disadvantage in operation, the broader the variation of pressure on the boiler the greater the injury to the joints and materials in the nature of a racking effect. The amount of blowdown allowable should be considered with reference to the pop pressure. It is considered good practice for the valve to be adjusted to close after blowing down not more than 4 lb. on biolers carrying 100 lb. per sq. in. gage; 6 lb. on boilers carrying 100 to 200 lb.; and not more than 8 lb. on boilers carrying over 200 lb. per sq. in. gage.

QUESTION [Resetting Valves to Change from Condensing to Non-condensing]—*What changes should be made in the valve setting of a Corliss engine if a change is made from running condensing to noncondensing?* A. H.

Answer—When running noncondensing the exhaust will be at higher pressure, and for smooth running of the engine the necessary cushioning on the exhaust can be obtained by less compression, obtained by closing the exhaust valves later in the stroke. On a single-eccentric engine, set the eccentric back on the shaft so that only the necessary compression of exhaust is obtained, and see that the exhaust-valve rods are short enough to obtain release before the end of the expansion stroke. Setting the eccentric backward will delay admission and the steam-valve rods will need to be shortened so the valves will have less lap and proper lead. With the wristplate hooked in gear, place the engine first on one center and then on the other and adjust the valve rods to suitable lengths for obtaining the proper lead and release. With a double-eccentric engine, setting of the steam eccentric and steam-valve gear needs to be no different than when running condensing, and for less compression of the exhaust, the exhaust eccentric and exhaust-valve gear would require the same attention as the valve gear of a single-eccentric engine, but omitting all consideration of the steam valves and their rods.

QUESTION [Setting Steam Valves of Duplex Pump]—*What is the method of setting the steam valves of a duplex pump?* J. R.

Answer—First, place the pistons of each side of the pump at the centers of their strokes by moving them first to contact with one head and then to the other, marking each position on the rods against the stuffing-box glands. Then, on each rod, place a mark midway between those marks and move the rod

until this central mark comes against the gland. When this is done on both sides, the pistons are in central positions, and the valve-motion rocker arms that connect with the crossheads should stand perpendicular to the piston rods. If for either side this is not the case with either crosshead it must be disconnected and fixed at such a position as to correct the and position of the arm. Now, remove the steam-chest covers place the slide valves centrally over the ports and adjust the lost motion of each valve so there will be the same clearance at each end. If, upon trial of the pump, a steam piston strikes the head of its cylinder, take up some of the lost motion; if the piston does not make a full stroke, give more lost motion.

QUESTION [Radial Valve Gear]—*What is a radial valve gear?* L. C. N.

Answer—The term radial valve gear is applied to reversing mechanisms having but one eccentric or equivalent crank motion.

QUESTION [Proportions of Link Valve Gear]—*For a link-motion reversing valve gear, what should be the length of eccentric rods and dimensions of the link, in terms of the throw of the eccentrics?* D. Y.

Answer—Ordinarily, the radius of the Stephenson link should be made equal to the length of the eccentric rods reckoned from the center of the eccentric. Without a rocker the radius may be greater and with a rocker it may be less, but deviation from the normal should not be enough to derange the leads. The eccentric rods should be twelve times the eccentricity of the eccentrics and the operative length of link should be four times the eccentricity or more, if space is available, although a skillful designer may employ the inequality introduced by short rods or a short link to adjust a link motion.

QUESTION [Slip of Link]—*In operation of a link-motion reversing valve gear, what is meant by the slip of the link?* L. B.

Answer—In operation the link and link block vibrate around different centers. Consequently, for each revolution of the crank of the engine the link slips back and forth a certain distance on its block, and this slippage of one on the other is called the slip of the link.

QUESTION [Point of Suspension of Stephenson Link]—*What determines the location of the point of suspension for a link of a Stephenson link-motion valve gear?* S. D.

Answer—The complex motion of the link and constrained motion of the block give rise to a slippage or rubbing of the link upon the block. The nearer the block is to the point of suspension the less there is of this slippage, and the link usually is suspended nearer the end connected with the forward

eccentric rod to reduce the wear for the more common working position. The slippage in backward gear is thereby increased; but when equal efficiency is required in both forward and backward gears, the link is suspended from the center.

QUESTION **[Direction of Setting Globe Valve]**—*What is the proper way to place a globe valve, to have the pressure under the disk or on top of the disk?* H. P.

Answer—For most situations, especially for large globe valves or for high pressures, it is preferable to have the pressure under the disk, as that method permits packing the valve when closed and prevents the accident of the valve stem pulling out of the disk when the valve is opened under pressure. Another advantage of having the pressure under the disk is that the direction of flow results in less barrsion of the valve seat. The principal advantages of having the pressure on top of the disk are that the pressure holds the disk to its seat; also that when the line is allowed to cool down with the valve closed, the lost motion between the disk and the spindle permits the disk to follow up the contraction of the valve body and when steam is again turned into the line—the valve is as tight as before, whereas with steam reintroduced on the seat side of the valve, there is leakage until the valve has become reheated and reexpanded to its former dimensions, and to close the valve down tight before it is warmed up makes it harder to open after the expansion has taken place.

QUESTION **[Advantage of Riding Cut Off Valve]**—*What is the advantage of a riding cutoff over a single automatic cutoff valve?* R. M.

Answer—For medium and slow speed engines, it is desirable to have uniform points of release and compression. With a single valve, the points of release and compression must vary with the point of cutoff. With a riding cutoff the main valve determines the lead, point of release and point of exhaust closure and these are constant, as the main valve is operated by a fixed eccentric. The duty of the riding cutoff valve is simply to close the ports in the main valve and effect the point of cutoff, as determined either by hand adjustment or by the governor, depending upon whether the engine is a throttling or an automatic cutoff engine.

QUESTION **[Replacing Corliss Valve Stems]**—*How would the key seats be laid off for new valve stems for a Corliss engine?* J. E. E.

Answer—With the old stems in place, uncover the valve bonnets and, with the center mark on the wristplate hub brought opposite to the center mark on the wristplate stub, make a mark on the end of each valve to register with a mark made on the valve seat. Then replace the old valve stems with the new ones, and with the wristplate at the central position and the marks on the ends of the valve registering with the

marks on the valve seats as before, the valve arms in place, scribe off the positions of the keyways from the key seats in the hubs of the valve arms. If the work of marking and cutting the keyways is accurately done, the valve setting will be the same as it was with the old valve stems.

QUESTION [Lap of Steam Valve on Long-Range Corliss Cutoff]—*How much lap has the steam valve of a long-range double-eccentric Corliss engine when the wristplate is in mid-position?* J. B.

Answer—When the wristplace is in midposition, the steam valve must have negative lap—that is, must be part way open. The lead is delayed by turning the eccentric backward from the 90-degree position, so the cutoff may act after the piston has completed one-half of its stroke.

QUESTION [Noisy Exhaust Valves]—*When our* 16 x 36-*in. Corliss engine is lightly loaded, there is a slapping noise from the exhaust valves. How can the trouble be prevented?* T. N. B.

Answer—Undoubtedly, steam-engine indicator diagrams would show that when carrying the light load, the cutoff is very short with expansion carried to a pressure below the atmosphere. When this occurs, the exhaust valves are forced from their seats by pressure of the atmosphere causing a slapping or clattering noise, as when an engine is being shut down. The remedy is to throttle the steam, or lower the boiler pressure, and obtain cutoff so late in the stroke that the steam will not be expanded below atmospheric pressure.

QUESTION [Determining Leakage of Piston and Valves]—*What is the best method of testing for leakage of the piston and valves of a Corliss engine?* B. H.

Answer—Let steam in one end of the cylinder and open the indicator cock at the other end and leakage of the piston can be detected by escape of steam. By closing the valves and opening both cocks, leakage of the valves can be detected. Leakage of the exhaust valve is shown by steam blowing out of the exhaust pipe when steam is admitted to the cylinder, with the indicator cocks and exhaust valves closed.

QUESTION [Exhaust Valves Must Not Have Lap for Compression]—*For obtaining compression on a single-eccentric Corliss engine, why is not lap given to the exhaust valves?* M. F. S.

Answer—If compression is obtained by giving lap to an exhaust valve, thus causing it to close the exhaust port before the piston reaches the end of the stroke, it would also make the valve later in opening and the steam would not be released at exhaust until after the piston had completed its stroke. Hence the proper method of obtaining compression is to advance the eccentric beyond the 90-degree position and thereby

obtain earlier release and earlier compression. Then to avoid excessive lead, that, is, to hold the steam valve closed during the period of compression or until it is desirable to have the live steam admitted, it becomes necessary to give lap to the steam valve.

QUESTION [Disadvantages of Excessive Lap of Corliss Valves]—*What are the disadvantages of excessive lap on the steam valves of a Corliss engine?* F. E. A.

Answer—Excessive lap reduces the rapidity and amount of movement of the valve at the point of opening, and also diminishes the range through which the engine will cut off.

QUESTION [Position of Exhaust Valves with Engine on Center]—*With a releasing valve gear, should the exhaust valve be open when the engine is on a center?* H. S.

Answer—In a cut-off engine, the steam should be released from the cylinder by having the exhaust valve open before the end of the stroke, to insure prompt discharge of the steam and prevent undue back pressure during the return stroke. Consequently, when the engine is on a center and the piston has arrived at one end of its stroke, the exhaust valve on the other end of the cylinder should be open.

QUESTION [Clattering of Balanced Valve With Circular Back]—*Our shaft governor engine is provided with a balanced valve that, judging from the large amount of steam used, probably leaks badly, and when the engine is running the valve makes a clattering noise such as made by most engines when started up or stopped. A short time after the valve and circular top-plate were planed and supplied with new packing rings, the noise became as bad as before. How can the trouble be remedied?* E. W.

Answer—The trouble complained of is common with the type of valve described, especially when the engine is operated with a heavier load than the load for which wear has caused shoulders to be formed on the steam-chest cover. For the greater valve travel with heavier loads, the circular edge of the top plate strikes these shoulders or travels over them with a clattering of the valve. The only remedy is to make frequent inspections of the surfaces and keep them true and parallel, and keep in order the springs that hold the valve-back against the steam-chest cover.

QUESTION [Inside Lap for "Cushioning on the Exhaust."]—*What is meant by inside lap of an engine valve and what is its effect in the operation of an engine?* R. G. L.

Answer—The lap of a valve is the distance its edge overlaps the port when the valve is in the middle of its travel. The term inside lap, when applied to a D slide valve, refers to the distance the exhaust edge of the valve overlaps the exhaust port when the valve is in the central position. Inside

or exhaust lap therefore results in closure of the exhaust port before the piston reaches the end of its stroke. The exhaust steam then remaining in the cylinder is thereby compressed and acts like a cushion for gradually arresting the piston as it approaches the end of the stroke, and hence the operation is commonly referred to as "cushioning on the exhaust."

QUESTION [Reason for Lap of Corliss Steam Valves] —*Why is lap given to the steam valves of a single-eccentric Corliss engine?* H. L.

Answer—To obtain early release after expansion and compression of the exhaust, it is necessary to advance the eccentric ahead of the 90-deg. position. With the eccentric advanced, all the valve events occur earlier in the stroke and, when the wristplate is in its central position, the steam valves must have sufficient lap to be closed long enough to prevent excessive lead.

QUESTION [Negative Lead]—*What is meant by negative lead, and how would a valve be set to give negative lead?* F. J. G.

Answer—Lead is the width of port opening when the crank is on dead center, and negative lead would be the amount the valve overlaps the admission edge of the port when the crank of the engine is on dead center. To set a valve with the same amount of negative lead at each end, turn the engine over and lengthen or shorten the valve rod, as may be necessary, to obtain the same travel of each end of the valve beyond the admission edges of the ports. Then place the engine on a center and set the eccentric in such a position that the valve overlaps the admission edge of the port for forward rotation by the amount of negative lead that is desired.

QUESTION [Angle of Advance with Negative Lap]—*When a D slide valve has negative steam lap, will the eccentric require positive or negative advance to close the steam port before the end of the stroke?* T. J. M.

Answer—With negative lap the valve has the port uncovered during more than one-half a revolution of the shaft, and if the eccentric should be set at 90 deg. with the crank, the port would be uncovered both at the beginning and at the end of the stroke of the piston. Hence to have the port covered before the end of the stroke the eccentric must have positive advance, that, is, must be set ahead of the 90-deg. position, which also would increase the lead or port opening at the beginning of the stroke and hasten all of the valve events.

QUESTION [Determining Lap and Lead of D-Valve]—*How can the steam lap and lead of a D-slide valve of an engine be determined without removing the steam-chest cover?* E. M.

Answer—Make a mark on the valve stem at the end of the stuffing-box gland when the stem is farthest in, and another

when it is farthest out of the stuffing-box, and halfway between these marks locate another mark to represent the mid travel of the valve. Then turn and block the engine wheel to such a position that steam will be just admitted to one end of the cylinder with the throttle valve open only a little, as shown by the escape of steam from the open cylinder pet-cock on the same end of the cylinder. Make a mark on the valve stem at the end of the stuffing-box gland and locate a similar mark with the valve just opening on the other end. The distance between these latter marks will be equal to the sum of steam laps of both ends of the valve, and the distance that each one is from the mark previously made to register the mid travel will be the amount of steam lap on corresponding ends of the D-valve. To determine the lead at either end, place the engine on a center and make a mark on the valve stem at the end of the stuffing-box. Note the distance of this mark from the mark made for registering the beginning of opening of the same end of the valve. If the engine would have to be turned backward to obatin the registration of the opening, then the distance so noted would be the amount of positive lead, but if it would have to be turned forward, the distance would be the negative lead, or amount the valve would lack of being open at the beginning of the stroke.

QUESTION [Lap and Negative Lead of Valve]—*What is the difference between lap and negative lead of a valve?* W. P. S.

Answer—The term lap is used to signify the distance the admission edge of a valve overlaps the admission edge of its port when the valve is in the middle of its travel. Lead is the distance the admission edge of the valve has uncovered the admission edge of the port when the piston is at the beginning of a stroke from the same end of the cylinder, and negative lead is the distance the admission edge of the valve overlaps the admission edge of the port when the piston is at the beginning of a stroke from the same end of the cylinder.

QUESTION [Reason for Greater Lap on Head End of Valve]—*Why should there be more lap of a D-slide valve for the head end than for the crank end of an engine cylinder?* W. R.

Answer—On account of the angularity of the connecting-rod there is less number of degrees rotation of the crankshaft and eccentric during the half-stroke in the head end than in the crank end of the cylinder. Therefore with equal laps, cutoff would occur later in the stroke from the head and, to more nearly eqaulize cutoff, more lap must be given to the valve on the head end. But with a direct-acting eccentric rod, or valve gear provided with a straight rocker, it is not practicable to add sufficient lap to entirely equalize the points of cutoff because the addition of lap on the head end reduces the lead on that end and lead is of more importance than equalization of cutoff. The unequal lap should not be more than will result in the greatest lead permissible for the crank end and the

least necessary for the head end. Equalization of both lead and cutoff can be obtained by employment of a properly designed bent rocker.

QUESTION [Steam Laps of D Slide Valve]—*If the length of a D slide valve is 25⅝ in. and the distance between the outside edges of the steam ports is 24 in., how much is the steam lap of the valve?* P. S. L.

Answer—The sum of the steam laps of each end of the valve would be $25\frac{5}{8} - 24 = 1\frac{5}{8}$ in., and when the laps are equal, each lap would be half of $1\frac{5}{8}$, or $\frac{13}{16}$ in.

Section X

Piping

QUESTION [Size of Pipe Line and Power for Pumping]—*What size of iron pipe line 1,200 ft. long would be suitable for delivery of 15 gal. of water per minute at 30-lb. pressure, pumped from a lake to an elevation of 85 ft? What would be the pressure on the pump from the line when not running, if the pump is placed 5 ft. above the lake, and what should be the horsepower of a motor for operation of the pump?* W. P. O.

Answer—With the discharge end of the line open to atmospheric pressure and the line filled with water, the static, or standing, pressure in the pipe at the pump would be $(85 - 5) \times 0.433 = 34.6$ lb. per sq. in.

Pumpage at the rate of 15 gal. per minute with a total lift of 85 ft. and discharge pressure of 30 lb. per sq.in., without allowance for power to overcome pipe friction, would require $\frac{[30 + (85 \times 0.433)]\ 15 \times 231}{12 \times 33{,}000}$ = about 0.6 net or water horsepower, and with a pump efficiency of 50 per cent, the required power would be $0.6 \div 0.50 = 1.2$ hp., neglecting pipe friction. For discharging 15 gal. per minute through 1,200 ft. of smooth, clean $1\frac{1}{4}$-in. iron or steel pipe, the pressure required for overcoming friction would be about 42 lb. per sq.in.; for $1\frac{1}{2}$-in. pipe, about 15 lb. per sq.-in.; and for 2-in. pipe, about 6 lb. per sq. in. With 50 per cent pump efficiency, each pound pressure, with discharge at the rate of 15 gal. per minute, would require $\frac{1 \times 15 \times 231}{12 \times 33{,}000 \times 0.50} = 0.0175$ hp. and the total power required using $1\frac{1}{4}$-in. pipe would be $(0.0175 \times 42) + 1.2 = 1.9$ hp.; using $1\frac{1}{2}$-in. pipe, $(0.0175 \times 15) + 1.2 = 1.5$ hp.; and using 2-in. pipe, $(0.0175 \times 6) + 1.2 = 1.3$ hp. The pressure and consequently the pump friction increases with use of a pipe line on account of an increase of roughness of the bore and reduction of diameter due to scale and corrosion. Therefore it would be advisable to provide a pipe not less than $1\frac{1}{2}$ in. in diameter and a motor of not less capacity than 2 brake horsepower.

QUESTION [Dry-Kiln Couplings]—*What is the difference between dry-kiln and common pipe couplings?* R. H.

Answer—In dry-kiln heating it is of importance that no leaks develop in the steam pipe or joints, or the contents of the kiln may be seriously damaged; and to insure against bad

joints, pipe couplings of extra-heavy weight and length are used, and couplings of the kind are known as dry-kiln couplings.

QUESTION [Copper-Wire Gaskets for Flange Joints] —*How are wire gaskets made and used for flange joints of steam pipe lines?* W. C.

Answer—For medium-sized flanges and moderate pressures wire gaskets usually are made of soft copper wire $\frac{1}{16}$ to $\frac{3}{16}$ in. in diameter bent and joined to the form of a circular ring of appropriate size for placing them between the flanges inside of the bolt circle. The rings are readily formed by cutting them from a coil made by winding the wire around a mandrel to the form of a close spring and the cut ends of the wire are made continuous by a soft-soldered butt joint.

QUESTION [Scaling-off of Poor Galvanizing]—*What preparation can be used as a coating to prevent galvanizing from scaling-off iron pipes?* J. J. C.

Answer—Ordinary galvanizing is a coating obtained by dipping the pipe or other article in a bath of molten zinc. If the zinc shells off, it is because the article when galvanized was not properly prepared or the temperatures were not suitable, and no preparation applied as a coating over the zinc that is already loosened could be of benefit for increasing its adhesiveness. A coating like paint or asphaltum would act as a binder until it became dry and brittle, but would be thrown off along with the galvanizing in places where the zinc has not taken good hold on the pipe. For improvement of appearance of the exterior and protection from rusting, remove all galvanizing possible to be rubbed off with a sharp coarse file and give the pipe a coat of aluminum paint.

QUESTION [Preference for Standard-Weight Pipe]—*For general power-plant work and for pressures upward of* 300 *lb. per sq.in., why is* 6- *to* 8-*in. standard pipe preferred to the extra-heavy?* W. E. A.

Answer—Standard-weight pipe is preferred to extra-heavy on account of greater flexibility. Extra-heavy pipie is considered to be more appropriate where pipes occupy inaccessible places or are especially exposed to corrosion. For 300 lb. pressure the factor of safety of standard pipe would be about $2\frac{1}{2}$, while extra-heavy would give a factor of safety of about $4\frac{1}{2}$. The selection with respect to safety must be left to the discretion of the designer.

QUESTION [Galvanizing Process]—*How are water fittings and other articles galvanized for the purpose of resisting corrosion?* B. N. M.

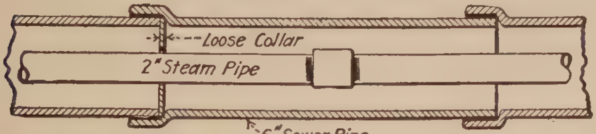
Answer—The articles to be galvanized must first be pickled in a dilute acid to remove or loosen all surface dirt and scale, and when removed from this bath, the articles are well

brushed with brooms or steel brushes and washed by dipping in fresh water or with clean water from a hose. They are then placed in a weak bath of muriatic acid to insure a clean metallic surface and taken directly from this bath to a flux bath that consists of a hot solution of sal ammoniac covered with beef tallow. From the fluxing bath the articles are lifted dripping and immediately lowered into the galvanizing bath. This consists of molten zinc, contained in a pot or iron tank surrounded by brick walls, with space enough between the walls and the tank to maintain a coke fire or heat of gas flame for keeping the zinc melted. The articles to be galvanized, suspended from wires or supported in a wire basket, are lowered in the zinc bath for a few minutes and then lifted out and allowed to cool.

QUESTION [Underground Conduit for Steam Pipe]—*It is desired to put down a 2-in. underground steam line about 200 ft. long for heating purposes. The pressure of steam delivered to the line will be 15 lb. What method of conduit construction is recommended where the soil is sand and clay?*

R. J. B.

Answer—A good conduit for 2-in. underground steam pipe can be made of 6-in. sewer pipe, as shown by the sketch. The



steam pipe is held centrally in the sewer pipe by iron washers that are like loose collars on the pipe. The outside diameter of the iron washers is the same as the outside diameter of the sewer pipe and the washers are placed at joints that are about six feet apart. The space around the steam pipe can be filled with loose insulating material or left as an air-space insulation The top and sides of the sewer-pipe joints can be made up with cement in the usual manner, but to take care of any water that may find its way into the conduit the under side of the joints should be left unsealed with provision for the subsoil drainage.

QUESTION [Cutting Pipe Threads on a Lathe]—*When cutting pipe threads on a lathe with taper attachment, should the lathe tool be set square with the taper or square with the lathe center?*

F. A. K.

Answer—The cutting tool carried by the taper attachment should be set square with the lathe center; that is to say, the tool is ground to cut a V-thread of 60 deg. and so set that each side of the finished thread will make an angle of 60 deg. with the lathe center, or what would be the same thing, 30 deg. each side of a line square with the lathe center.

Section XI

Gages

Section XI

Gages

QUESTION [Trouble with Glass-Gage Breakage]—*Much trouble is experienced with breakage of water-gage glasses on one of our boilers. What may be the cause and the remedy?*

W. R.

Answer—The durability of a water-gage depends largely on the method of packing the glands and on proper alignment of the gage-glass fittings. Test the alignment by placing through the fittings a round bar turned to the size of the bushings. A new glass should have a diameter not less than $^3/_{64}$ in. less than the bore of the gland nut and bush, and the glass tube should have at least $\frac{1}{8}$ in. end play to allow for expansion, observing, however, that the glass is long enough to prevent the packing from being forced over the end of the glass and cause obstruction of the glass-tube ends or the passages of the fittings. Tighten up the top and bottom gland nuts alternately. With the fittings in good alignment and glands properly packed, sufficiently tight joints should be secured from screwing up the gland nuts by hand, or at most by a gentle touch with a wrench. If the glass tube cannot be turned with the comparative ease by the finger and thumb after the gland nuts have been set pressure tight, it may be taken for granted that the glands are improperly packed or that the fittings are out of line.

QUESTION [Use of Loop or Siphon for Connecting Steam Gage]—*Why is a loop used in connecting a steam gage to a boiler?*

R. N.

Answer—A loop or siphon is placed in the connection between the boiler and the gage to trap water of condensation in the connecting pipe, so that steam will not enter the gage tube and affect its stiffness.

QUESTION [Trying Glass-Gage Connections]—*How often should the glass water gage of a boiler be blown out?*

M. D. T.

Answer—Glass gages should be blown out at least once a day, and as much oftener as may be found necessary to make sure they are in proper working order, and when this is done the operator should be careful to see that the cocks are open or the gage will not indicate correctly. If the lower connection is open and the top one closed, the boiler pressure will force the water too high in the glass, and if the bottom connection

is closed and the top one open, the steam entering at the top will be condensed and gradually show a higher water line than actually in the boiler. Of course if both connections are closed, the gage will not indicate changes of the water level and high or low water may result without the knowledge of the fireman.

QUESTION [Intensity of Pressure Exerted by Water and Mercury]—*What would be the indication of a pressure gage placed at the foot of a* 6-*in. standpipe* 100 *ft. high when the pipe is filled with water and when filled with mercury?*
J. J. S.

Answer—When the liquids are at the temperature of 62 deg. F., the intensity of pressure of a column of water 1 ft. high is 0.433 lb. per sq. in., and that of a column of mercury 1 ft. high is 5.892 lb. per sq. in. A pressure gage so placed that the center of the gage is level with the foot of the water column 100 ft. high would indicate $0.433 \times 100 = 43.3$ lb. per aq. in., and the gage so connected that its center would be level with the foot of the mercury column 100 ft. high would indicate $5.892 \times 100 = 589.2$ lb. pressure per square inch. In each case the pressure shown by the gage would depend on the height of the liquid and would be independent of the diameter of the standpipe.

QUESTION [Obtaining Correct Gage Pressure]—*We have some low-pressure steam gages located about* 10 *ft. below the steam lines, that indicate about* 5 *lb. above the pressure of the steam. How can the gages be connected to show the correct pressure?*
E. W.

Answer—When a correct gage is placed below the pipe, the pressure indicated will be the pressure in the pipe plus the pressure due to the height of water column in the connection, measured from the bottom of the steam main to the center of the gage. Assuming that the temperature and density of the water in the connecting pipe are constant, the plus quantity will be the same for all pressures and the pressure indicated when there is only atmospheric pressure in the steam pipe and with the gage pipe filled with water will be the amount to be deducted from all reading to obtain the gage pressure in the steam pipe If it is desired to obtain correct readings at sight, without making the subtraction, reset the gage pointer or supply the gage with an extra pointer, to indicate zero on the dial when the connection is filled with water and the gage pressure in the steam pipe is 0.

QUESTION [Convex and Concave Bumped Heads]—*How are bumped heads of steel plate named with respect to the manner in which they are set in drums, and what is their relative strength with respect to the side receiving pressure?* F. A.

Answer—Bumped or dished heads are called convex or concave according to the form of the side that is presented

outside of the drum or other vessel in which they are used. In good practice the concave heads of drums or other pressure vessels that have the pressure on the convex side of the heads are allowed a maximum working pressure equal to 60 per cent. of that allowed for dished heads of the same dimensions with the pressure on the concave side.

QUESTION [Physical Properties of Materials]—*What is meant by the terms "elastic limit," "modulus of elasticity," "permanent set," "ultimate strength," "percentage of elongation" and "reduction of cross-section" with reference to materials of construction?* R. N. B.

Answer—The elastic limit is a more or less definite value of the unit of stress beyond which, as the stress is increased, there is a greater relative deformation than the increase of stress. "Permanent set" is used to represent the lasting deformations produced by stresses greater than the elastic limit. "Modulus of eleasticity" is a term used to express the ratio of the unit stress to the deformations per unit length within the elastic limit. The total stress under which a body fails is called its "ultimate strength." The ratio of the total elongation of the specimen to its original length is called the "percentage of elongation." For calculations of percentages of elongation measurements are usually taken between marks originally 8 inches apart. "Reductoin of cross-section" is usually expressed as a ratio of the smallest cross-sectional area after rupture to the original cross-sectional area.

QUESTION [Irregular Water Level in Glass Gage]—*What causes the water in a gage-glass to continually rise and fall?* E. E. R.

Answer—Fluctuations of the gage-glass water-level may be caused by a surge or wave of the water in the boiler, due to circulation, or it may result from irregular formation or release of steam in the parts of the boiler to which the water column is connected.

QUESTION [Isothermal and Adiabatic Expansion]—*What is meant by the terms isothermal expansion and adiabatic expansion?* E. P.

Answer—When a gas expands at a constant temperature it is said to expand isothermally. In order that the temperature will remain constant it is necessary for heat to be supplied from outside to compensate for the heat that is converted into work. Isothermal expansion may be approximated in a steam engine by means of a steam jacket around the cylinder.

A gas expanding in a non-conducting cylinder would be said to expand adiabatically (that is, "no heat passing through") and its temperature would fall during the expansion, since there would be no heat supplied to take the place of heat that was converted into work. True adiabatic

expansion is probably never realized in practice on account of the difficulty of securing the variations of temperatures that should exist at the different stages of the expansion; but adiabatic expansion is closely approximated when the expansion occurs suddenly, as in steam turbine, for there is then little time for the conduction of heat.

QUESTION [Capacity of Cylindrical Tanks]—*What is a short rule for computing the number of gallons contained by a cylindrical tank 72 in. in diameter by 18 ft. long, with flat heads; and what allowance should be made when the heads are bumped?*

$$\frac{(72\times72\times0.7854)\times(18\times12)}{231}=$$

$$(72)^2\times18\ \text{ft.}\times0.0408=3{,}807.13\ \text{gal.}$$

W. R. J.

Answer—The cubical content in American gallons of the tank with flat heads is given by the computation: that is, where d = diameter of the tank in inches and L = length in feet, the general formula for capacity of a cylindrical tank with flat heads would be given by the formula,

$$Gallons\ capacity=d^2\times L\times\ 0.0408$$

Where h = the height of a bumped head in inches and d = its diameter in inches, the volume of a bumped head is given in gallons by the expression:

$$[(\frac{h}{2}\times 0.7854\,d^2)+(h^3\times0.5236)]\div231=$$

$$\frac{h^3\times0.7854d^2}{2\times231}+(h^3\times0.00226).$$

Hence two bumped heads would contain the same number of gallons as a cylinder of diameter d and length h, $+2\ (h^3\times 0.00226)$ gal., and the formula for complete capacity, including both bumped heads would be

$$Gallons\ capacity=$$

$$(d^2\times\ length\ of\ sides\ in\ feet+\frac{h}{12})\times0.0408+2\ (h^3\times0.00226)$$

Tanks for holding liquids under atmospheric pressure usu. lly are provided with heads that are bumped so little that the value obtained for $2\ (h^3\times0.00226)$ is only a small percentage of the total tank capacity, and when there are two equal bumped heads the capacity is given close enough for most practical purposes by the formula

Number of gallons = dia. in in. × *dia. in in.* × *length of side plus height of one head in feet* × 0.0408.

QUESTION [The "Locomobile" Power Plant]—*Why is the locomobile type of power plant so called, and what are the reasons for its superior economy?*

R. L.

Answer—The name "locomobile" came from the fact that as originally made these power plants were mounted on wheels

and intended for portable use. The high economics are due to the precautions taken in their design to guard against initial condensation and to minimize loss of heat in the flue gases and the exhaust steam. The more economical plants are operated compound condensing. Steam generated in the boiler is passed through a superheater suspended in the smoke box with the direction of steam flow so arranged that the hottest steam is heated by the hottest gases. The steam then passes through a pipe within the flue to the high-pressure cylinder, which is jacketed by the hot flue gases. From the high-pressure cylinder the steam passes to a reheater receiver contained in the smoke box and is thence admitted to the low-pressure cylinder, exhausting through a feed-water heater to the atmosphere or to a condenser. The boiler-feed pump, and condenser pump if used, are driven directly from the main engine.

QUESTION [Protection of Steel Smoke Flue from Rusting]—*We have considerable trouble from interior rusting of sheet-steel smoke flues that connect boiler uptakes with the chimney. How can the corrosion be prevented?* W. B. R.

Answer—The best protective covering for the purpose is a wash of portland cement and water mixed to a consistency of cold-water paint and applied with a stiffbrush after the surfaces to be covered have been thoroughly scraped and cleaned.

QUESTION [Form of Chimney Flues]—*Why are smoke-stacks made larger at the bottom than at the top?* E. E. R.

Answer—Brick chimneys and self-sustaining steel stacks are generally made larger at the bottom to increase their stability against wind pressure and sometimes only for architectural effect. On account of loss of temperature of the gases in their ascent through a chimney flue, they are somewhat denser at the top of the chimney, and enlargement of the flue at the bottom may have some advantage in obtaining greater uniformity of velocity of the gases. But for chimney flues of ordinary proportions, such an advantage is negligible and the draft capacity may be considered as equivalent to that of a flue of uniform size and of cross-section equal to the smallest part.

QUESTION [Chimney Capacity for Greater Height and Smaller Flue]—*A stack having a flue* 36 *in. in diameter is* 78 *ft. high. How much will the capacity be increased if the height is increased to* 100 *ft. with the size of flue reduced to* 34 *in. in diameter at the top, and what will be the increase in force of draft?* J. N L.

Answer—For the same temperature of chimney gases the force or pressure of the draft at the base of the chimney will be directly as the height, or $(100-78)\times100\div78$ = about 28 per cent. greater, thereby permitting use of a lower grade of fuel. The discharge capacity is directly as the

square root of the height and about proportion to the area, so that the boiler capacity with the proposed dimensions would be to the present capacity as $34 \times 34 \sqrt{100}$ to $36 \times 36 \sqrt{78}$, or only about 1 per cent. greater.

QUESTION [Casehardening Pins of Roller Expander] —*How can the pins of a roller expander be casehardened?*

F. A. K.

Answer—A quick process of casehardening small articles, like expander pins, is to plunge the part at bright-cherry heat into a box containing cyanide of potash crystals and then quench the piece at a dull-red heat. Cyanide is a deadly poison and should be handled accordingly. The operations of casehardening should be performed in the open air, the operator taking his position on the windward side of the cyanide, with care not to inhale or expose his eyes to any gases or vapors formed in carrying out the process.

QUESTION [Return Trap Connections]—*We are installing some machinery that requires a supply of live steam and will have return traps for returning the condensation to the boiler, carrying* 85 *lb. pressure. One of the machines is to be supplied with steam reduced to the pressure of* 30 *lb. gage. How should the returns and trap be arranged and connected?* F. K.

Answer—All returns connected to a trap or receiver should be at the same pressure; otherwise those of higher pressure will prevent the trap from simultaneously receiving returns of lower pressure. Systems of returns that are at different pressures will require separate traps. The return pipes should be sloped so they will be drained perfectly clear of condensation and discharged by gravity to the trap with the trap placed four feet or more above the water line of the boiler. It is well to place an automatic air valve on the return line near the trap and also provide a pet-cock for testing whether the returns are being emptied. When the trap becomes filled with return water, the mechanism automatically admits live steam from the boiler for siphoning the water into the boiler, but cannot do so unless the receiver is elevated sufficiently for obtaining the necessary head of discharge water for overcoming friction of the trap discharge pipe and fittings. There should be a separate return connection to the boiler, with check valve or connection so arranged that the discharge will not be affected by pulsations of a boiler-feed pump. The inlets and outlets for admission and discharge of return water, and for admission of live steam and discharge of displacing steam, are usually described in directions issued by the manufacturer or are indicated on the trap. If the water inlet of the trap is not provided with a check valve, one should be placed in the return line near the trap. Most return traps are provided with a small outlet for discharge of the displacement steam. This should be provided with a stop valve and pipe

connection for discharging the steam in the boiler ashpit or at some other point where a small quantity of air steam or water discharge from the trap would not be objectionable.

QUESTION [Height of Raising Water by Return Trap]—*How high will a return steam trap raise water?* D. M.

Answer—The height depends on the density of the water, the pressure of steam acting on the water for expelling it from the trap, the back pressure or pressure ahead of the water, and the pressure required for overcoming friction of pipes, valves and fittings. The density of water at 62 deg. F. is 62.36 lb. per cu. ft., and as a column of water at that temperature and one foot high would exert a pressure of 62.36 ÷ 144 = 0.433 lb. per sq. in. for each pound pressure of steam in excess of back pressure and pressure required for overcoming friction, the water could be raised to a height of 1 ÷ 0.433. = 2.309 ft. At 200 deg. F. water weighs 60.07 lb. per cu. ft., and without allowance for back pressure and pipe friction, for each pound pressure of steam acting on the surface of the water it could be raised to a height of 1 ÷ (60.07 ÷ 144) ÷ 60.07 = 2.397 feet.

Section XII

Belting

Section XII

Belting

QUESTION [Relative Capacities of Rubber and Leather Belting]—*What is the relative transmissive capacity of rubber and leather belting?* C. W. C.

Answer—When a proper weight of duck is used in rubber belting, four-ply rubber belting is taken to have the same transmission capacity as good quality, single-thick leather belting of the same width; six-ply rubber the same capacity as double thick leather belting and eight-ply rubber the same capacity as triple thick leather belting.

QUESTION [Width for Machine Belt]—*What width of single leather belt would be suitable for transmission of* $3\frac{1}{4}$ hp. *to a machine that is provided with an iron receiving pulley* 12 *in. in diameter, to run* 275 *r. p. m.?* W. L. F.

Answer—Each inch of good-quality single leather belt running on an iron pulley 30 in. in diameter and making 100 r.p.m., or 785 fet. per min., may be depended upon for transmission of 1 hp., and for the same stress of belt per inch of width, transmission of $3\frac{1}{4}$ hp. to a 12-in. pulley running 275 r.p.m. would require a belt width of

$$\frac{100 \times 30}{275 \times 12} \times 3\tfrac{1}{4} = 2.9, \text{ or practically 3 inches.}$$

QUESTION [Appropriate Width for Double Belt]—*What width of open double leather belt would be suitable for transmitting* 82 *hp. from a* 72-*in. diameter driving pulley to a* 26-*in. diameter iron receiving pulley that is to make* 325 *r.p.m. with pulleys space* 17 *ft.* 9 *in. horizontal centers?* R. P. H.

Answer—For reasonable service and durability it would be suitable to provide good-quality double leather belting on the basis of capacity for transmission of 1.8 as much as single belting, and rate the transmission appropriate per inch width of single belting as 1 hp. per 785 ft. travel per min., or 1 hp. per inch width of single belt on a 30-in. diameter pulley making 100 r.p.m. For the conditions stated this would require a double belt $\frac{82 \times 30 \times 100}{1.8 \times 26 \times 325}$ = about 16 in. wide.

QUESTION [Depolarized Belts]—*Pumps, driven through depolarized belts, for transmitting gasoline, have been ordered for one of our plants. How is depolarization of belts accomplished?* S. R.

Answer—Belts running at high speeds in dry places take on a high static electrical charge. The sparks caused by the discharge from such belts become very dangerous where gasoline fumes or other highly explosive fumes are present. The usual method of overcoming static is to place one or more metal combs in close proximity to the belt and connect them to ground. Pieces of small chain or wire may be used for the same purpose. Driving these pumps direct would seem to be a much better practice, as this would eliminate any trouble that might arise from the use of a belt.

QUESTION [**Advantage of Inclined over Vertical Belt Drives**]—*Why will a belt on an angle of about* 45 *deg. transmit more power than a vertical drive?* F. A. S.

Answer—When a belt is placed over a pair of pulleys, both sides of the belt may be assumed to have the same tension, but while transmitting power the belt stretches from greater tension on the driving or tight side, which immediately results

FIG. 1

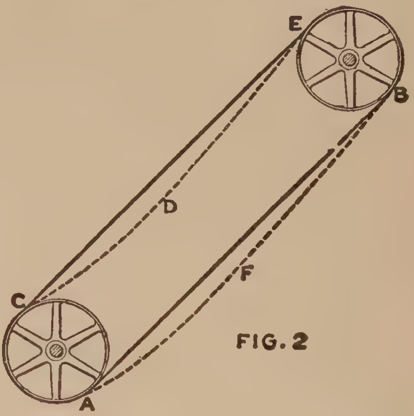


FIG. 2

in less tension on the slack side. When the sides are vertical, as in Fig. 1, the slackening may be sufficient to permit the belt to fall away from the lower pulley as indicated by the dotted line, or at least reduce the pressure and friction between the belt and the face of the lower pulley and this condition is aggravated by the weight of the belt. When the belt is inclined as shown in Fig. 2, increase of tension on the driving side reduces the tension on the slack side. When the lower side *AB* is the driving or tight side, the slack side, indicated by the dotted line *CDE*, becomes wrapped around more of the lower pulley; and when the uppser side *CE* is the tight side, the slack side assumes a position like *AFB* and remains wrapped around nearly as much of the lower pulley. In either case the belt remains supported by the lower pulley and there is assistance of the weight of the belt to produce pressure against the

face of the lower pulley. Hence for the same initial tension a belt is capable of transmitting more power than when the drive is vertical.

QUESTION [Lengths of Splices for Leather Belts]—*What is the rule for the length of splices for leather belting?*
W. R. K.

Answer—For belts up to 10 in. wide, make the splices 10 in. long. Belts that are 10 to 18 in. wide should have the splices as long as the belts are wide. Eighteen inches is the greatest length required for the splice of a double belt.

QUESTION [Belt Thickness Influences Speed Ratio of Pulleys]—*Should not a 36-in. pulley drive a 12-in. pulley at three times the speed of the former?* R. F.

Answer—If the pulleys operate as friction pulleys running in contact without slip, the 12-in. pulley will be driven at three times the speed of the 36-in. pulley. But if separated and the driving is done by a belt without slippage, the small pulley will be driven at something less than three times the speed of the large one on account of the thickness of the belt. In making calculations of speeds without slippage, an amount equal to the thickness of the belt should be added to the diameter of each pulley.

QUESTION [Effective Pull of Belt]—*What is meant by "belt pull" and how is it computed? Is belt pull the same as belt stress? What would be the belt pull in driving a machine that requires 20 hp. where the belt is 2,325 ft. per minute?* W. N. J.

Answer—The term "belt pull" is commonly employed to signify the effective pull or difference in tensions of the tight side and slack side of a belt when transmitting power. The actual pull or tension that tends to draw the belt apart is the tensile stress and the intensity of that stress usually is expresed in pounds per square inch of cross sectional area of the belt.

For transmission of 20 hp., or $33,000 \times 20 = 660,000$ ft.-lb. per minute, and for a belt speed of 2,325 ft. per min. the effective pull would be $660,000 \div 2,325 = 283.0$ lb. For transmission of any given power with a given speed of belt, the maximum tensile stress, or actual pull coming on the tight side of the belt, depends on the required effective pull, the tension under which the belt was put on pulleys and elasticity of the belt material.

QUESTION [Cleaning Leather Belts of Oil]—*How can machine oil be removed from leather belts without injury to the material?* H. C. R.

Answer—Remove all gum or glaze from the surfaces by scraping them with a broad putty knife or similar scraper with rounded corners so they will not cut or scar the belt surface and then, with the belt stretched out on a floor or bench,

clean first one side of the belt and then the other by a spreading of dry sawdust swept off with a clean stiff broom, Then place the belt in a warm place for a few days, packed in clean, fine sawdust, powdered chalk or fuller's earth for absorption of the oil by capillary action.

QUESTION [Power Transmission for Difference in Belt Tension]—*What would be the difference in tension on the tight and loose side of a belt traveling* 1200 *ft. per min. to transmit* 6 hp.; *and what number of horsepower would be transmitted with the same difference in tension for a belt speed of* 400 *ft. per min.?*
M. D.

Answer—Transmission of 6 hp., or $6 \times 33,000 = 198,000$ ft.-lb. per min., at a speed of 1,200 ft. per min. would require a difference of tension of $198,000 \div 1200 = 165$ lb., and a belt speed of 400 ft. per min. with the same difference of tension would transmit $\frac{400}{1200}$ of 6, or 2 horsepower.

QUESTION [Improving Flexibility of Leather Belting] —*What is the best method of improving the flexibility of a new leather belt is not mineral oil good for the purpose?* W.K.S.

Answer—Mineral oil is a solvent of albuminous substances and therefore is detrimental to the natural cementing substance of leather. For improvement of the flexibility, remove the glaze from both sides of the belt with a damp cloth and give both side a liberal coat of lukewarm neatsfoot oil, rubbed off dry after standing 10 to 12 hours. A new leather belt should be thus treated several times a month and afterward kept moderately "stuffed" with the oil to prevent stretching and cracking from alternate absorption of moisture and drying out with changing conditions of the atmosphere.

QUESTION [Results of Belt Failures with Blocked Governors]—*With a blocked engine governor, what would happen if the main driving belt should break or if the governor belt broke or came off?* W. D. T.

Answer—If the main belt should break, the load would be removed and the engine would race and perhaps be wrecked. With the governor belt broken or run off, the governor would supply the amount of steam corresponding to the blocked position, without power to increase the supply for an increase of the load so that a load greater than that capable of being carried by the blocked position of the governor would cause the engine to slow down, but any less load would permit the engine to race and very likely to become wrecked.

Section XIII

Oils

QUESTION [Testing Cylinder Oil for Tarry Ingredients]—*How can a cylinder oil be tested for tarry residue?*

P. M.

Answer—Dissolve one part of the oil in about twenty parts of gasoline. Allow a sample to stand for an hour in a clean test-tube and the tar or other insoluble matter will collect on the bottom of the test-tube.

QUESTION [Removing Burnt and Gummy Oil]—*What will clean burnt oil from the top of a steam chest and cylinder head of an engine?* W. R. T.

Answer—Burnt and gummy oil can be removed with a strong lye water, made of about a pound of potash to a half bucket of water. Remove the cylinder head and lay it down with the face covered over for ten to fifteen hours with waste or cloths saturated with the lye water and apply the wash in the same manner to the top of the steam chest. This will soften the dirt so it can be readily scraped off the surfaces, which can then be cleaned up with sandpaper and brightened with oil and emery cloth.

QUESTION [Moisture in Transformer Oil]—*What is a simple method of testing transformer oil for moisture?* W. E. S.

Answer—Anhydrous copper sulphate when placed in transformer oil will give a bluish tint if moisture is present in the oil. If some small pieces of hot iron or nails are dropped into a sample of the oil the presence of moisture will be indicated by the hissing sound produced. Before applying either one of these tests by the uninitiated, it will be advisable to make the tests on samples that are known to contain moisture and others that are known to be "dry," so as to become familiar with the reactions under both conditions.

QUESTION [Removing Oil from New Boilers]—*How can new boilers be boiled out to get rid of considerable oil and grease in them? Will oil and grease left inside of the boilers by the boilermakers have any injurious deffect in operation?*

L. E. G

Answer—A simple method of removing grease and oil is to boil off the grease with about one-tenth pound of soda ash to a horsepower of boiler capacity. Pulverized soda ash can be introduced through any opening above the water level in the boiler when the gage pressure is zero. After the soda ash is

added, fill the boiler nearly full of water and, with the safety valve raised so there cannot be accumulation of pressure above that of the atmosphere, hold the water to the boiling point with a slow fire for about 24 hours, and then draw off the boiler and thoroughly wash the interior with clean water and wash out the safety valve and other valves and connections, to remove all gumming or other traces of the soda ash. More than one such boiling off and washing may be necessary for removal of unusual quantities of oil and grease. Generally, the amount left by the makers is not enough to cause boilers to become burnt, but if not boiled off, the oil is likely to cause priming and trouble from leaks that may be started by the oil or prevented by its presence from becoming rust tight.

QUESTION [Gravity, Flash and Fire Test of Lubricating Oils]—*What effect have gravity and flash or fire points on the lubricating value of oils?* *B. T.*

Answer—Flash and fire points of an oil are important considerations where over-heating by bearings might result in ignition of waste or other combustible materials, and the gravity may be an index for identification of a particular kind of oil, but none of these properties can be regarded as indicative of lubricating value. The real test is the analysis which comes from observation of the effect on the friction load.

QUESTION [Preventing Siphonage of Oil]—*We are having trouble with leads on a number of starting compensators siphoning oil from the tanks. What method can be recommended to remedy this difficulty?* A. A. F.

Answer—The siphoning of oil along the leads of transformers or compensators may be prevented by first removing the insulation from each lead above the highest oil level for about two inches. Then sweat the strands of the leads solidly together so that there will be no openings between them for the oil to pass through. After this is done, insulate the leads with treated linen tape, wrapping the tape very tightly around the conductor, and paint each layer with insulating varnish while wrapping. Allow the tape to extend about one inch down over the insulation on the conductor.

QUESTION [Starting-Compensator Oil]—*What is the best grade of oil to use in starting compensators employed on 220-volt induction motors ranging in size from 10 to 75 hp. Would a poor grade of oil cause the contacts to burn?* M. C. R.

Answer—In compensators employed for starting squirrel cage motors a mineral oil having a high flash point should be used; it should be free from any trace of alkali or acid and ave a low factor of evaporation. Under operating conditions the oil should not form deposits in the tank. One of the surest ways of securing the proper grade of oil is to obtain it from the manufacturers of the compensators. The design of the contacts, operating conditions and application of the com-

pensator in general have a much greater influence on the burning of the contacts than the grade of oil. If the contacts are designed to give a rubbing make-and-break and the compensator is not abused either through improper handling or misapplication, little trouble should be experienced with burning and pitting, although a certain amount of this will occur under the best of conditions. In general the oil should be changed when it turns dark in color and sediment forms in the bottom of the tank.

QUESTION [Removing Grease from New Boilers]—*How can oil and grease be removed from a new boiler before it is put in service?* J. S. A.

Answer—Place in the boiler about half a pound of soda ash per horsepower capacity and, with the boiler filled with clean water to the highest gage cock, start a slow fire. After holding the water to the boiling point for about twelve hours, allow the fire to die out and the boiler to cool slowly, after which drain the boiler and then open it and wash out thoroughly.

QUESTION [Carbon in Lubricating Oil] — *After refiltering the lubricating oil in a Diesel engine, I find a lot of carbon particles still in the oil. Can this be remedied?* R. G. N.

Answer—It is practically impossible to remove all the free carbon from the oil. A small amount will do no damage.

QUESTION [Inexactness of Relations of Viscosimeter Tests]—*What are the respective relations of tests of the viscosity of oil obtained by Saybolt, Engler, Redwood and Tagliabue viscosimeters, and how are they obtained?* H. S.

Answer—In each of the instruments the viscosity value is based upon the number of seconds of time taken by a set amount of oil to flow through an orifice that has been calibrated for the time required for the orifice to discharge the same quantity of water. There is no generally accepted tests for viscosity as various types of instruments are used, and the results with different instruments will often vary considerably. None of the types have orifices of eaxctly the same size. Unless the names of the designer and maker of the instrument used and the amount and temperature of oil used are stated, the results are almost meaningless. Therefore it is impossible to translate the readings of one instrument to that of another with any accuracy.

QUESTION [Treatment of Hot Bearings]—*What is a good treatment for a hot main bearing that must be kept running?* R. D. B.

Answer—One of the best remedies for a hotbox or main bearing is to loosen up the bearing all around, then give it a generous supply of cylinder oil through the oil holes, or feed the bearing with a mixture of about one measure of graphite and ten of heavy engine oil. If heating does not disappear,

at first opportunity clean the bearing and try to run it down smooth with a light load while the bearing is kept flooded with a very thin mixture of flowers of sulphur in clean water.

QUESTION [Grease More Dangerous to Burning New Boilers]—*Are new boiler tubes more likely to be burned from deposit of oil or grease than old ones?* G. T.

Answer—It has been found that clean new flues and other heating surfaces are more affected by oil than old surfaces, especially where the latter are coated with an appreciable amount of scale. Many cases of burning of new boilers have been known where the thickness of the oil or grease coating has probably been less than one-thousandth of an inch.

QUESTION [Cylinder Oils for Compound Engines]—*Are different kinds of cylinder lubricating oils necessary for use in the high- and low-pressure cylinders of a compound engine?* R. C. N.

Answer—For obtaining good lubrication it is sometimes necessary to use different grades of oil. The drop of pressure in the high-pressure cylinder causes condensation and the supply of steam to the low-pressure cylinder will be wet, with a tendency to wash the film of oil off the surface of the cylinder and make lubrication more difficult. The oil used for the low-pressure cylinder should be of a quality that will readily combine with moisture and cling to the cylinder walls, and the oil obtainable for that purpose may not be suitable for the high-pressure cylinder.

QUESTION [Oiliness and Viscosity of Lubricant]—*What is the difference between oiliness and viscosity of a lubricating oil, and how do these respective properties aid lubrication?* A. D. B.

Answer—A lubricant is said to be oily when it is in nature of a limpid greasy liquid that readily spreads and attaches itself to the bearing surfaces without chemically combining with their material, and keeps the surfaces separated with reduction of the friction without gumming or cutting of the surfaces. Viscosity, or "body," of a lubricating oil is the measure of its fluidity and therefore an index of the cohesiveness of its particles. It is advantageous to lubrication if no greater than necessary to give strength of cohesiveness and adhesiveness required to hold the oil together in a film of sufficient thickness to prevent engagement of the irregularities of the opposing lubricated surfaces and to prevent the lubricant being pressed from between the surfaces.

Section XIV

Gas and Oil Engines

QUESTION [Diesel-Engine Output]—*What fuel consumption per kilowatt-hour can one expect from a Diesel engine at full load in ordinary operation?*

S. C. H.

Answer—A Diesel oil engine in fair operating condition will deliver a kilowatt-hour at the expense of 0.65 lb. of fuel oil.

QUESTION [Noise and Breakage of Diesel-Engine Piston Rings]—*We have a two-cylinder* 100-*hp. two-stroke-cycle semi-Diesel operating at* 257 *r.p.m. The rings on the pistons of this engine have persisted in "snapping" even immediately after rings and pistons have been cleaned. No lubrication difficulties have been encountered, and almost no ring sticking has been experienced. The engine has been operated approximately* 4,000 *hours. No breakage of the rings has occurred except at the ends where they are pinned to prevent turning on the piston. The snapping noise seems to occur from the same cause that breaks the ends of the rings, and there is no evidence of excessive wear either on the ring seats or cylinder walls. Can you suggest the cause and remedy for the trouble?* G. E. Z.

Answer—This snapping is caused by a too light fit of the rings in the grooves. The piston, in moving down on the power stroke, is thrust against one side of the cylinder; the ings are then pushed back so that they are flush on this side of the piston and protrude on the other. On the upstroke the tight rings remain in the same position, holding the piston against the cylinder wall. Upon the injection and firing of the charge, which takes place before top dead center, the cylinder pressure throws the piston against the cylinder wall opposite to where it is bearing; the rings are then forced back into the groove with a sharp snap. By filing the edges of the rings to give more clearance, the trouble should disappear.

QUESTION [Effect of Elevation on Horsepower of Gas Engine]—*How does the altitude of the place where a gas engine is used effect the horsepower of the engine?* R. W.

Answer—The power of a gas engine varies with the atmospheric pressure and consequently with change in elevation. If b = barometric pressure at sea level, e = barometric pres-

sure at the elevation where the engine is used, hp = the horse, power at sea level, and hp_e = the horsepower at the elevation-

then $hp_e = \frac{e}{b} hp$.

QUESTION [Removing Scale from Water Jacket]—*How can a gas-engine water jacket be cleaned of scale deposited out of the cooling-water?* D. L. S.

Answer—The scale deposited out of most cooling waters can be removed by a solution of commercial muriatic acid and an equal quantity of water. Care should be taken that the solvent action of the acid is not continued to the extent of injuriously corroding the walls of the water jacket or other parts of the engine. After application of the acid solution, the parts that have come in contact with it should be thoroughly washed with clean water.

QUESTION [Horsepower of Gasoline Engine]—*What horsepower may be expected from our four-cylinder four-stroke-cycle gasoline engine having cylinders* 6 *in. diameter by* 6 *in. stroke*, 500 *r.p.m.?* H. N. G.

Answer—Engines of this type are usually rated at 1,000 ft. piston speed and the horsepower is calculated by the formula

$$Hp = \frac{D^2 \times No.\ cylinders}{2.5}$$

Since the engine has four 6-in. diameter cylinders, at 1,000 ft. piston speed, which with a 6-in. stroke is equivalent to 1,000 r.p.m., the horsepower $= \frac{36 \times 4}{2.5} = 57.6$, and at 500 r.p.m. the engine would be rated at $\frac{57.6 \times 500}{1,000} = 28.8$ horsepower.

QUESTION [Reduction of Baumé Reading to Pounds of Oil Per Gallon]—*Knowing the Baumé reading of a fuel oil or distillate, how are the pounds per gallon estimated?* C. M.

Answer—The weight per gallon will be equal to the weight of a gallon of pure water multiplied by the specific gravity or weight of the oil as compared with the weight of an equal volume of pure water. One U. S. gallon of pure water at the temperature of 62 deg. F. weighs 8.3356 lb. and one Imp. gallon of pure water at the same temperature weighs 10 lb. To convert degrees Baumé to specific gravity, the following formulas are used:

For liquids heavier than water, sp.gr. = 145 ÷ (145 − deg. Bé.).

For liquids lighter than water, sp.gr. = 140 ÷ (130 + deg. Bé.).

Hence for an oil lighter than water, if the reading per the Baumé hydrometer is 21 deg. Bé. the specific gravity would be 140 ÷ (130 + 21 deg. Bé.) = 0.927, and the weight would be 8.3356 × 0.927 = 7.727 lb. per U. S. gallon.

QUESTION [Efficiency of Internal-Combustion Engine at High Speed]—*Why is it that with the same engine the efficiency at 300 r.p.m. is greater than at 900 r.p.m.? If, as usually conceded, the frictional loss is constant regardless of the speed, it would seem that at the higher speeds and larger power output the efficiency should increase.* J. H. R.

Answer—The discrepancy lies in the poorer scavenging of the cylinder at the high speeds, which lowers the thermal efficiency. In addition, at high speeds the combustion, which is by no means instantaneous at any speed, is carried on during the entire power stroke. This means that the heat, instead of being added at the highest temperature, is added at a falling temperature, giving a decreased efficiency. In addition, part of the fuel may be exhausted in an unburnt condition. On the other hand, with proper design, the high-speed machines may be more efficient than a slow-speed engine.

QUESTION [Cause of Back Firing]—*What is the cause of back firing in a gas engine?* W. A.

Answer—This is usually caused by a slow burning of the gas mixture, the flame continuing until the inlet valve opens, whereupon the incoming charge is ignited. At times a highly heated part of the engine will ignite the gas as it flows into the cylinder, blowing back through the gasmixing valve or carburetor. Also, a leaky inlet valve may cause back firing.

QUESTION [Lubricating Oil for Diesel Engines]—*Is it good practice to use the same lubricating oil for the piston and the main bearings in a semi-Diesel engine?* H. C.

Answer—Most manufacturers provide a single mechanical lubricator for supplying the same lubricant to all parts of the engine. In choosing an oil for such general use, it must not be too heavy for the bearings or too light for the piston. Undoubtedly, a separate oil for each duty would be better.

QUESTION [Higher Thermal Efficiency from Lean Mixture]—*Why does a lean mixture in an internal-combustion engine give a higher thermal efficiency than a richer mixture?* M. S. T.

Answer—Considering the Otto or constant-volume-cycle

engine, the thermal efficiency is $E = 1 - \frac{1}{r^{K-1}}$, where r is ratio of initial, or suction volume to the final, or compression, volume, and K is the ratio of the specific heats at constant pressure and constant volume. In an existing engine r, the volume ratio, is fixed and the only variable is the specific heat ratio K.

If nothing but pure air was in the cylinder, K would be 1.408. Gases or gasoline vapors have a lower specific heat ratio, and as the mixture gets richer, K for the entire mass decreases and the fraction $\frac{1}{r^{K-1}}$ becomes larger, making the efficiency lower. The leaner the mixture the nearer K approaches 1.408 and the greater is the thermal efficiency.

Viewed in another way, C_p the specific heat at constant pressure is made up of the specific heat at constant volume plus the work done or $C_p = \text{Work} + C_v$ To make the ratio $\frac{C_p}{C_v}$ large C_v must be small for a given heat addition. This means that upon ignition of a charge containing a given amount of heat the maximum temperature will be higher, since the B.t.u. added per degree is small, that is, C_v is small. In expansion the steepness of the curve is proportional to K, or with a small C_v the curve will be steep and the final exhaust temperature will be low. The temperature drop will then be greater than with a more gradual curvature of the expansion line. Since the efficiency depends upon the maximum and exhaust temperatures, the efficiency is greatest with K approaching 1.408, or where a lean mixture is used.

QUESTION [An Oil-Engine Compressor Valve Sticks]—*We find that the valves of the compressor on our Diesel engine cake with carbon when the engine runs on low loads for any length of time. What is the cause?* P. J. K.

Answer—Since this does not occur on full loads, it is evidently due to high-temperature conditions on low loads. The reason the temperature of the discharge is high on low loads is due to throttling the suction air. This lowers the suction pressure and increases the ratio of discharge to suction pressure. The temperatures vary with the pressure ratio, and on low loads it is possible for the temperature of the last stage to be above the vaporizing point of the lubricating oil. As a result the light portions of the oil are vaporized, leaving a gummy residue which settles on the valves. To remedy this, try a high-test oil or run with the suction line open and let the excess air escape through a relief valve.

QUESTION [Explosion Temperature in Internal-Combustion Engine]—*What is the highest temperature at-*

ained in the cylinder of an internal-combustion engine at the time of explosion? J. J. L.

Answer—The maximum temperature that occurs in an internal-combustion engine at the time of explosion varies with the compression that is carried in the engine and also with the amount of fuel added. Since in an Otto engine the amount of fuel added is limited by the amount of air that is taken in on the suction stroke and also by the permissible maximum pressure at explosion, the temperature will have a range of from 2,500 to 3,500 deg. F. In the ordinary gas engine, carrying from 80 to 90 lb. compression, the maximum temperature will be in the neighborhood of 2,700 deg. F. On the other hand, with oil engines where the compression pressure is carried much higher, the maximum temperature often is as high as 3,500 to 3,600 deg. F. Theoretically, the temperature should reach about 4,500 deg. F., figuring the amount that is normally added to the air charge. However, there is a very rapid radiation of heat during the period of combustion, and this serves to keep the temperature much below the theoretical value.

QUESTION [Waste Heat Recoverable from Diesel Engine]—*How much heat can be obtained from an exhaust heater on an oil engine?* G. N.

Answer—If part of the jacket cooling water is sent through an exhaust heater, over 35 per cent of the heat in the fuel supplied to the engine can be recovered with a final water temperature of 212 degrees.

Simple Tests of Mineral Oils

QUESTION [Simple Tests of Mineral Oils]—*What simple engine-room tests can be made for determining the quality of engine and machine oils?* R. N. C.

Answer—For determining whether an oil is well refined, fil a clean bottle about half full of the oil and heat slowly over an open flame, until vapor appears above the oil surface, and maintain that temperature for 15 minutes. If well refined, thel oil will darken in color, but will remain perfectly clear and free from sediment after standing for 24 hours; but if the oil has been poorly refined, it will turn black and, after standing, a black deposit will appear.

To test for tarry and other residues, place a small quantity of the oil in a test tube along with about 20 times the quantity of 85 Bé. gasoline. Allow the mixture to stand for an hour, and the tar and other insoluble matter present will collect at the bottom of the test tube.

A well-refined oil should show no trace of sulphuric acid. A simple test for acid is made by partly immersing a piece of

highly polished copper in the oil for about 24 hours. If a trace of acid is present, the immersed surface of the polished copper will be made dull. The simplest test for sulphuric acid is to wash a sample of the oil thoroughly with warm water and test the water with neutral litmus paper. If even a faint reddish tint is shown on the paper, the oil should be rejected.

QUESTION [Use of Crude Oil as Fuel]—*Why is not crude petroleum used for boiler fuel?* W. R. F.

Answer—Crude oil contains many highly volatile constituents that can be distilled off and have high market value in the forms of gasoline and allied distillates. The presence of these volatile constituents makes the oil more dangerous to store or handle, as combustible vapors are given off in large quantities at comparatively low temperatures and the mixtures formed with the atmosphere are highly explosive. Hence, the material generally marketed as fuel oil is a residuum left after the more volatile constituents of the crude oil have been distilled off, and it has nearly the same properties as the crude oil without giving off the dangerous vapors at so low a temperature.

QUESTION [Specific Gravity Corresponding to Baumé Degrees]—*What are the specific gravity values corresponding to gravities of petroleum oils quoted in Baumé degrees?* L. P.

Answer—The corresponding values of the Baumé scale and specific gravity within limits most used in connection with petroleum are as follows:

Baumé Deg.	Spec. Grav.	Baumé Deg.	Spec. Grav.
20	0.9333	34	0.8536
22	0.9210	36	0.8433
24	0.9090	38	0.8333
26	0.8974	40	0.8235
28	0.8860	42	0.8139
30	0.8750	44	0.8045
32	0.8641	46	0.7954

QUESTION [Conversion of Baumé Degrees to Specific Gravity]—*How is the specific gravity of a liquid estimated from the number of degrees Baumé?* G. A. W.

Answer—To convert degrees Baumé into specific gravity the following formulas are used:

For liquids lighter than water:

$$Sp.\,gr. = \frac{140}{130 + deg.\,Be.}$$

For liquids heavier than water:

$$Sp.\,gr. = \frac{145}{145 - deg.\,Be.}$$

QUESTION [Sulphur in Fuel Oil for Internal-Combustion Engines]—*For oil engines is it advisable to use a fuel oil carrying considerable sulphur? We can purchase such an oil for about* 25 *per cent less than an oil of the same gravity but having only* 0.5 *per cent of sulphur.* W. R. S.

Answer—With modern designs of engines, there is no reason why oil carrying as high as 3 per cent sulphur should not be used. The sulphur undoubtedly will unite with the water vapor formed during the combustion of the fuel if the temperature in the exhaust is below the saturation temperature. This can be avoided by carrying the exhaust line fairly warm.

QUESTION [Proportion of Air for Explosive Oil Vapor]—*Is it true that an oil vapor will not be explosive unless it is mixed with an exact amount of air?* W. L. M.

Answer—If a volatile oil is mixed with air, it will not explode if the proportion of the volatile be very small nor will explosion occur if the proportion of oil is large. The explosion range with gasoline is from 2 to 4 per cent. of fuel vapor to 100 per cent. of air. However, explosions frequently occur with exceedingly rich mixtures. This is due to the outer edges of the mixture being of the right proportion, and on igniting will so diffuse the remainder through the air that the entire oil charge will explode.

QUESTION [Flashing Point of Fuel Oil]—*What is meant by the flash point of a fuel oil, and what do the "open" and "closed" tests consist of?* W. T. D.

Answer—The flashing point of an oil is that temperature to which the oil must be heated, at a specified rate, to cause it to give off a vapor which will ignite when mixed with air and exposed to a flame near the surface of the oil. The "open test" consists in heating some of the oil, in which the bulb of a thermometer is placed, in a small open metallic cup, a porcelain crucible embedded in sand or some equivalent contrivance, heated by application of a gas flame or other source of heat that is under the control of the experimenter so that the temperature of the oil increases at the rate of 10 deg. F. per minute. A small flame from a taper or gas tube is passed across the surface of the oil for every five degrees rise in temperature until the vapor ignites or flashes. The temperature at which this flash is first observed is taken as the flashing point of the oil. The open-cup method is beset by uncertainties. The temperature at which the first flash is obtained depends on the presence or absence of air currents, the actual rate of heating, the size and shape of the vessel containing the sample, the distance of the flame from the oil surface and the skill of the experimenter.

The closed test, as the name implies, is performed with apparatus from which air currents are excluded and in which the oil and vapor are stirred and the heat applied may be so

regulated that when the temperature nears the flash point, the rise in temperature is so reduced that the testing flame may be quickly applied to the surface of the oil every two degrees rise of its temperature until the flashing point is reached.

Open-cup testing usually gives 5 to 25 degrees higher flash point than closed-cup testing, and the presence of much moisture causes irregular and incorrect results by either method of testing.

QUESTION [**Testing for Water and Dirt in Fuel Oil**]—*What is a good method of determining the amount of water and dirt in fuel oil?* W. T.

Answer—Mix a small quantity of the oil in a glass test tube with an equal quantity of gasoline and allow the mixture to stand in a warm place for 24 hours, when the dirt and water will settle to the bottom.

QUESTION [**Otto and Diesel Cycle Efficiencies**]—*Considering the Otto and Diesel cycles for internal-combustion engines, which is the more efficient?* D. H. W.

Answer—The efficiency of the Otto cycle may be written,

$$e = 1 - \frac{1}{r^{k-1}}$$

when r is the compression volume ratio and k the ratio of specific heats, about 1.35. The Diesel efficiency is,

$$e = 1 - \frac{1}{r^{k-1}} \left(\frac{p^k - 1}{k\,(p-1)} \right)$$ when p is the cutoff ratio.

The term in the brackets is always more than unity. Consequently, for equal volume ratios the Otto cycle is more efficient. Practical considerations prevent the employment of the Diesel ratio in the Otto-cycle engines and the Diesel engine as built is the more efficient.

QUESTION [**Explosions in Diesel-Engine Air Lines**]—*In a Diesel engine the safety plugs on the injection air lines blow out repeatedly. What causes this?* R. N.

Answer—It is apparent that explosions occur in the air line. These explosions can be caused in two ways. If the fuel valve spring is very weak, the explosion pressure in the cylinder will force the valve open, allowing the flame to blow through the air line, igniting any deposits of lubrication oil in the line. Also, if the injection air leaves the compressor very hot, it may ignite any lubrication oil deposited in the air pipes.

QUESTION [**Live-Steam Coils for Oil-Storage Tanks**]—*What is the objection to heating fuel oil in a storage tank with a pipe coil supplied with live steam?* S. S. D.

Answer—The objection raised against the use of live-steam heating coils is that the temperature of the oil might be raised above the flash point and thereby liberate inflammable vapor.

QUESTION [Maximum Efficiency of Diesel Engine]—*Is there a limit to the efficiency obtainable from a Diesel engine?* B. W.

Answer—The maximum possible thermal efficiency when operating between the usual temperature limits is 59 per cent, this being the so-called "air cycle" efficiency. With oil as fuel the maximum indicated efficiency is around 49 per cent.

QUESTION [Smoky Exhaust of High-Compression Oil Engine]—*What would cause a two-cycle high-compression hot-cup oil engine to have a smoky exhaust and preignition?* C. M.

Answer—Both troubles result from imperfect combustion. With a good quality of oil, imperfect combustion and the resulting smoky exhaust may be due to excessive water cooling of the cylinder head and hot cup. So called preignition is pre-explosion of hydrocarbonaceous material that has been left over by imperfect combustion of a preceding stroke.

QUESTION [Smoky Exhaust of Diesel Engine]—*Will low injection air cause my Diesel engine to smoke?* G. M.

Answer—It will, especially if the engine is fully loaded. Air pressure below 50 atmospheres will often cause a smoky exhaust.

QUESTION [Oil in Semi-Diesel Engine Exhaust]—*Why does my 50 horsepower semi-Diesel have a continual stream of oil dripping out of the exhaust, and how can this be remedied?* E. S.

Answer—Not knowing all the conditions, it is natural to assume that you are running your engine with the hot ball too cold or the fuel nozzle leaks oil into the engine late in the stroke. The remedy is apparent.

QUESTION [Independent Exhaust Lines for Diesel Engines]—*We have two Diesel engines, each of 200 hp. Cannot both exhaust into a common exhaust pit relieved by a single exhaust pipe extended up alongside of our main building?* S. F.

Answer—It is preferable to have an independent exhaust pit and exhaust line for each engine. When the exhausts are connected, the products of combustion backed up from an engine in use have a bad corrosive effect on one that happens to be standing idle, besides being a source of interference when it is desired to inspect, adjust or repair one of the engines while the other is in use.

QUESTION [Sloping Combustion Line of Diesel Diagram]—*I cannot get an indicator diagram from my Diesel having a horizontal combustion line. Why?* G. L. N.

Answer—Although at one time all Diesels gave a horizontal combustion line on the diagram, at present many Diesels give

a sloping line. Actually, this increases the engine efficiency and eliminates the difficulties with the cam profile encountered where attempting to secure combustion at constant pressure.

QUESTION [Diesel Hard to Start]—*In starting a Diesel engine using* 26-*deg. oil, the cylinders do not fire until the flywheel has turned over a dozen or more times. How can this be improved?* D. B.

Answer—Evidently the cylinders are worn, losing compression until warmed up by making several strokes. As an emergency, use kerosene to start, since it fires at a lower temperature.

QUESTION [Oil Engine Cannot Be Started with High Air Pressure]—*When it is attempted to start our Diesel engine with the air-injection pressure around* 900 *lb. per square inch in the fuel valve, the engine fails to fire, but with pressures of* 600 *to* 700 *lb. the engine fires promptly. Why is this?* R. M. A.

Answer—The cylinders of your engine are probably worn, causing a fairly low compression pressure and a low final temperature. If the injection air pressure is high, the chilling action from expansion through the nozzle lowers the cylinder temperature below the ignition point of the fuel oil.

QUESTION [Welding Fractured Crankshaft of Oil Engine]—*Can a fractured oil-engine crankshaft be welded successfully?* C. B. H.

Answer—Electric welding of crankshafts has been uniformly successful. This does not mean that an inexperienced man can weld successfully. Such work should be turned over to firms who are experts in the work. If the weld is ground and polished, it is easy to determine if it is perfect.

QUESTION [Cracked Parts of "Dry Type" Semi-Diesel Engine]—*What would be the cause of cracked bridges of the air and exhaust ports and piston of a "dry type" semi-Diesel oil engine?* M. G.

Answer—Cracking of the parts mentioned could be caused by excessively high exhaust temperatures as a result of lime scaling of the water jacket, or a leaking atomizer, or from overloading the engine.

QUESTION [Repairing Piston Fire Cracks]—*I have a* 100-*hp. oil engine that has a fire crack in the piston head* 8 *in. long. Should I use cast or wrought brass to "sew" it?* J. M.. T.

Answer—It is necessary to use wrought brass since the cast brass will not admit of good riveting and will not stand up under the high temperature conditions.

QUESTION [**Imperfect Diesel Indicator Diagrams**]—*With new fuel cam noses put on a Diesel, it is found that the indicator diagrams are very sharp, showing no admission lines. Is this caused by the new cam noses?* H. C. C.

Answer—The new noses may give too great a lift to the fuel needle valve, allowing the oil charge to blow into the cylinder at once. Try giving the fuel rocker arm less clearance, which would cause the needle to open earlier and slower. Lowering the injection-air pressure often flattens the card. Your timing may be too early; if so, the maximum pressure will be above normal.

QUESTION [**Repeated Breakage of Gas-Engine Crankshaft**]—*We are having considerable trouble from repeated breakage of the crankshaft of an 80-hp. producer-gas engine. May not the trouble come from improper adjustment or poor lubrication of the bearings and crank boxes?* S. S.

Answer—Repeated breakages are more likely due to shocks from preignition and impacts to which internal-combustion engines are subjected. A crankshaft usually breaks after the material has become crystallized, and when a break has occurred, it may be taken for granted that more or less crystallization has taken place throughout the whole crankshaft material. The original structure can be nearly recovered by heat treatment, and the whole crankshaft should be so treated occasionally—at least whenever any part is repaired by rewelding.

QUESTION [**Cleaning Water Jacket of Gas Engine**]—*What solution can be used for cleaning the scale from the water jacket of a gas engine?* R. M.

Answer—The scale usually consists of lime deposited out of the circulating water and can be removed by a solution of one part commercial muriatic acid and four parts of water. As soon as the solution has done its work, the water jacket should be thoroughly washed out with clean water to prevent unnecessary corrosion.

QUESTION [**Oil Engine Fails to Synchronize**]—*Why do my oil engines refuse to synchronize?* R. N.

Answer—In many cases the coefficient of fluctuation in an oil engine is such that the variation in the flywheel velocity is entirely too great for the generators to stay in step. Using heavier flywheels or a squirrel-cage winding on the pole face will eliminate the trouble.

QUESTION [**Gumming of Compressor Valves**]—*What causes Diesel air-compressor valves to gum?* P. R. C.

Answer—Gumming results from using too much lubricating oil. Two drops per minute per cylinder is ample for the average-sized compressor.

QUESTION [Carbonizing of Diesel-Engine Cylinder After Repairs]—*I find that the cylinders of my Diesel carbonize badly. What is the reason? I recently put in new crankpin bearings. Would this cause the trouble?* J. M. C.

Answer—In putting in the new bearings, you probably failed to replace sufficient shims between box androd end. Low compression resulted, which would produce soot or carbon.

QUESTION [Guarantee Test of Oil Engine]—*How would a test be made of the guaranteed oil consumption of a 40-hp. oil engine running a 25-kw. 220-volt direct-current generator used for lighting?* H. P. K.

Answer—To make the test conclusive the economy and regulation should be determined as nearly as possible for the different loads, speeds and other conditions specified in the guarantee. If constant loads of the given magnitude are not obtainable supplying power to the regular load circuit, the desired loads can be obtained by use of a temporary water-rheostat such as illustrated and described on pages 180 and 181 of Aug. 7, 1917, issue of *Power*. If the generator is not to be included in the guarantee, it will be necessary to take into account the efficiency of the generator at the different test loads. These data are generally obtainable of the manufacturer. For reliable results each test should run for a period of at least three hours, noting the weight, kind, source, and analysis of oil used; and readings should be taken at five-minute intervals of the speed of the engine and of the volts and amperes of electrical output. For any test period the average (volts × amperes) ÷ 1000 will be the average kilowatt output, and the number of pounds of oil used per hour divided by the average kilowatts generated will be the oil consumption per kilowatt-hour.

QUESTION [Amount to Set Up Bearings]—*How much should the main bearings of a gas engine be set up to allow for lubrication and quiet running?* J. O. B.

Answer—The amount cannot be generally stated, as in any case it depends on the alignment, material, workmanship, condition of the journal and bearing, viscosity and wearing properties of the lubricating oil, uniformity of supply and distribution of the lubricant, the surface speed and pressure and kind and amount of dirt likely to find its way between the bearing surfaces. Gage the approximate thickness of shims required when the caps are set down hard on the journals with pieces of soft lead wire compressed between the shim surfaces. Try out the setting of the caps with ample thickness of shims and gradually reduce the thickness until the bearings show a tendency to cut or heat.

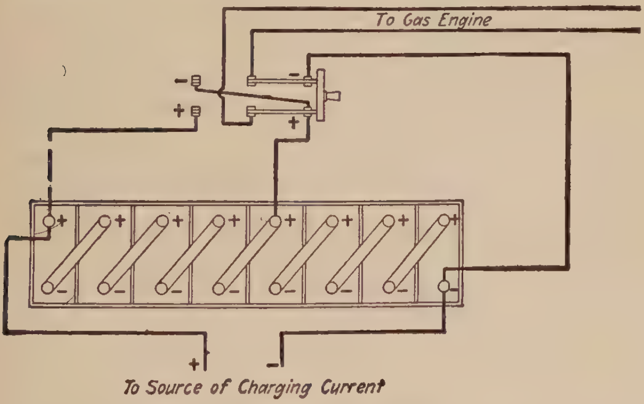
QUESTION [Alternators in Parallel]—*We have two crude-oil engines, each rated at 25 hp., which we want to belt to*

alternating-current generators of whatever size the engines will drive. It is desired to operate the alternators in parallel. Can this be done? Would there by any advantage in belting both engines to a lineshaft and driving the alternators from this shaft using clutches on the engine pulleys? W. L. H.

Answer—It would not be good practice to belt the two oil engines to the same lineshaft. This has been tried a number of times but without success. There is no reason why an alternating-current generator cannot be belted to each oil engine and have the generators operate in parallel. Alternating-current generators, to operate in parallel successfully when driven by these oil engines, will require squirrel-cage windings in the pole faces.

QUESTION [Battery for Sparking Gas Engine]—*We have a 16-volt storage battery that I would like to use for ignition purposes. The gas engine requires only about 8-volt current. Can a resistance be connected in series with the battery, so as to cut the voltage down to about 8 at the coil terminals, and use it to spark the engine? The battery is charged from a low-voltage generator.* L. K.

Answer—It is doubtful if satisfactory operation can be obtained by connecting a resistance in series with the storage battery. A better scheme would be to connect the battery



through a two-pole double-throw switch, as shown in the figure, so that four of the cells can be used at a time for ignition purposes. With this scheme the battery may be charged at any time, even when part of it is being used to ignite the engine.

QUESTION [Crediting Value of Waste Heat Supplied to Boilers]—*Our boiler plant is completely equipped for oil burning, and some of the boilers are operated most of the time with heat obtained from waste gases of metallurgical furnaces;*

but at intervals it is necessary to operate the whole plant with fuel oil. How can the monthly service of waste gases be credited to the furnaces in terms of barrels of fuel oil saved? R. G.

Answer—To evaluate the service, it would be necessary to determine the total number of pounds of water evaporated per month by the boilers when supplied only with waste gases, reduce the evaporation to equivalent evaporation from and at 212 deg. F., and ascertain the equivalent evaporation from and at 212 deg. F. for the same boilers when operated as nearly as possible with the same average output, using oil as fuel. The evaporation per month using waste gases divided by the evaporation obtained per barrel of fuel oil will give the number of barrels of fuel oil to be credited.

QUESTION [**Color for Natural-Gas Flame**]—*In the use of a natural-gas burner should the flame show bluish or straw color?* E. M.

Answer—A white or straw-colored flame is indicative of incomplete combustion, usually due to poor mixture of air or insufficient air supply, and although there may be greater capacity, there is less economy and more soot than from a blue flame. While a blue flame is representative of more perfect combustion, it may be less economical than a white or yellow flame if there is an excessive air supply. For best results the air supply should be just sufficient to prevent a white or straw-colored flame.

QUESTION [**Changing from Natural Gas to Producer Gas**]—*The natural gas for our gas engine will shortly be shut off. Is it possible to use producer gas without any great change to the engine? What will be the fuel consumption?* G. E.

Answer—Producer gas may be used, and all that is necessary is to install a producer. The engine will require no alteration beyond a possible change in the compression and in the proportions of gas and air passing through the throttle valve. Any of the well-known producers will deliver enough gas from anthracite coal to allow the engine to develop a brake horsepower hour on $1\frac{1}{4}$ lb. of anthracite pea or buckwheat coal.

INDEX

BELTING

BOILERS

ELECTRICITY

FUEL and FURNACES

GAGES

OIL and GAS

OIL

PIPING

PUMPS

REFRIGERATION

STEAM ENGINES

STEAM TURBINES

VALVES